Combustion

August 1961

High Temperature Water vs. Steam
for space heat, process heat, air conditioning

C-E makes *both* types of equipment. Here are the facts:

High temperature water systems are coming into wider use — not only for space heat and process heat applications, but for use with absorption air conditioning equipment as well. Customers have ordered some 80 High Temperature Water Boilers (Type HCC) for these purposes, with plant capacities ranging from 10 million to 300 million Btu/hr, and operating at water pressures up to 500 psi and temperatures to 470F, or higher.

The "Thermal Flywheel"

With a steam system, boilers often have to be operated constantly at full load. Wide load swings can cause loss of steam pressure and considerable lag before sufficient steam can be supplied. But because a high temperature water system has a far greater heat storage capacity than a steam system at the same pressure, even sudden large demands will not drain heat from consuming units. Thus fluctuating demand does not immediately affect boiler loads. Boilers operate at a more constant rate which improves the efficiency of the entire system. In industrial applications, temperature of ovens, tanks, calender rolls, heating platens, etc., can be controlled within close limits, which often results in improved product quality and increased production.

Savings with High Temperature Water

While capital costs for steam generators and high temperature water boilers of equivalent capacity are roughly the same, a system using a C-E Hot Water Boiler can usually save from 10 to 20 percent in operating and maintenance costs:

1. No steam traps. This means substantial savings in initial cost, plus further savings by eliminating trap maintenance. Steam losses due to normal trap operation are also eliminated;

2. No blowdown required. This represents a considerable loss in steam systems, reflected in increased fuel costs;

3. No pressure reducing valves. Initial valve cost is reduced and losses from improper valve regulation are prevented;

4. No pipe grading required. The system is always under pump pressure. Piping can be carried over machinery, in roof trusses, up or down grades —
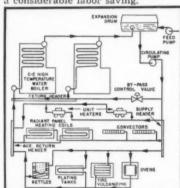
even underground. Pipe life expectancy is increased many times since corrosion is minimized in a closed oxygen-free system;

5. No condensate return lines. And no investment outlays for corrosion prevention; cost of replacing wet returns is eliminated;

6. No expensive feedwater treatment. Since only small quantities of make-up water are needed, treatment can be handled on a batch system basis.

Versatility of High Temperature Water

For heating units such as space heaters, radiant panels, etc., high temperature water may be used directly. For domestic heaters, ovens, tanks, vulcanizing equipment, etc., it can be used indirectly — as low temperature hot water or as steam — produced in suitable heat exchangers.
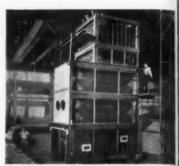
In addition, high temperature water is well-adapted to use in the coils of absorption refrigeration units — for cooling, as well as heating, multi-building applications. Absorption equipment can be located in each building and can operate without the supervision of licensed engineers, at a considerable labor saving.

A 30 million Btu Type HCC boiler partially shop assembled, showing compactness of the unit. Upon addition of refractory and welded casing, boiler will be lifted by its lugs onto a flat car for shipment.

fired by oil, gas, coal or any combination of these fuels. They operate with low pressure loss, provide complete control of system and boiler circulation, and offer optimum arrangement of heating surfaces. All circuits are drainable and headers are accessible from outside the boiler casing. With no baffles in the boiler, there are no areas where soot can accumulate. Symmetrical arrangement of heating surfaces permits a rugged, gas-tight casing construction which can be pressure fired and is entirely suitable for outdoor installations.

The standardized design of these units results in lower first cost, shorter delivery time, lower cost of ownership. Smaller sizes are completely shop assembled, while larger sizes are shipped in component assemblies, to reduce field erection costs.

You be the Judge

Because most operators, engineers and contractors are more familiar with steam than with hot water systems, we have gone to some length to enumerate the major differences between the types. Naturally, needs vary, and the eventual choice of a heating system should be made only after a thorough investigation of all factors.

Our engineers will be pleased to discuss either hot water or steam units, or both, with you and your consultants — impartially and with no obligation. For further details on high temperature water boilers, write for catalog HCC-2.

A simplified schematic arrangement of typical high temperature water applications in industrial or commercial plants. Shown are the Type HCC boiler, expansion tank, circulating pump and actual heat consuming units. Heat exchangers for conversion to low pressure hot water are also used frequently in these systems.

The C-E High Temperature Water Boiler

C-E High Temperature Water Boilers are designed around the same controlled circulation principle used by C-E in many of the world's largest steam generation units. They can be

Combustion

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

volume 33   number 2   **August 1961**

COVER PHOTO

The spray formation close to the nozzle of a swirl type nozzle is dramatically shown here.

Installation time at Leidy Station cut 25% with prejacketed J-M METAL-ON® pipe insulation!

a
butterfly
at
heart....

At the heart of every Graver **MONOVALVE** Filter is one butterfly valve—the one valve so important that it gives the filter its name. With a flip of its "wings" it performs the vital function of directing water to the filter beds, or sending it past them.

The **MONOVALVE**, independent as a butterfly, requires no operator attention since its performance is completely automatic, eliminating much of the chance of human error.

But the **MONOVALVE** Filter is no butterfly when it comes to work. It is a fully automatic gravity filter, with special plastic sand-retaining "Partilok" strainers. Backwash is initiated automatically by pressure drop across the filter bed, so the **MONO-VALVE** can never be overloaded. Because it stores its own backwash water, the **MONOVALVE** assures uniform and complete backwashing every time. And, no water is wasted in developing a vacuum here—the only water that goes to the sewer is that which is actually used to backwash the bed.

Accelerated life tests conducted with the **MONOVALVE** Filter have shown no effects of wear or fatigue after the equivalent of twenty years of service.

The versatile **MONOVALVE** Filter can be obtained in single or multi-compartment units. Write for Bulletin WC-130, to discover its applications for your filtration problems.

Industrial Department I-333

GRAVER WATER
CONDITIONING CO.
216 West 14th Street, New York 11, New York

ONE OF THE
UNION
U
TANK CAR
COMPANIES

© Graver Water Conditioning Co., 1961

C-E's VU-60

SOME DESIGN HIGHLIGHTS OF THE VU-60

VU-60
TEST RESULTS

Modular increments allow assembly of custom designs from standardized components; boiler can be proportioned for best combustion, heat absorption, gas flow — even where space conditions are difficult.

Completely self-cased, the VU-60 is an all-welded pressure-tight envelope composed of modular panels of finned tubes; fewer field welds cut erection time.

Insulation and preformed lagging go directly over tube panels, thus reducing non-working weight per pound of steam generated.

Characteristic of Combustion's VU-60. This superheater curve indicates a temperature stability throughout entire load range.

ALL TYPES OF STEAM GENERATING, FUEL BURNING AND RELATED EQUIPMENT; NUCLEAR REACTORS; PAPER

GIVES YOU A CUSTOM-DESIGNED BOILER
BASED ON STANDARDIZED MODULAR COMPONENTS

Because of its unique modular concept, C-E's new VU-60 offers unusual flexibility and can be designed to meet unusual or difficult space conditions.

The VU-60 is a completely self-cased boiler. Finned tubes, shop-welded into modular panels, make up the exterior walls of the unit. The fins joining the panels are field-welded to make a pressure-tight envelope. Insulation is applied directly over these walls and is covered by pre-formed, field-installed metal lagging. The need for bulky or heavy non-working structural members is kept to a minimum.

The VU-60 can be designed to meet your most exacting steam needs. It is easy to install, economical and dependable in operation, completely accessible and functional. Why not get full information on how this new concept can fulfill your specialized requirements?

COMBUSTION C-E ENGINEERING

Copes-Vulcan automatic boiler control at Hammermill Paper Company

Gives good results on steam generator fired with hogged bark and coal

Here was an unusual problem of boiler control. Two solid fuels—hogged bark and bituminous coal—were to be burned in combination. The Riley steam generator, with nominal rating of 206,000 pounds per hour at 675 psig and 760 degrees F., was to deliver power and process steam to the same main header as all other existing boilers. Low steam costs and high boiler availability were a goal of the design and operating engineers.

Copes-Vulcan combustion control has given high operating efficiency by close control of excess air. It has minimized fuel-bed disturbances by keeping the flow of air through the bed at the lowest practical rate. It has handled intermittent flows of waste fuel without incident. It has required no manual adjustments for variations in fuel moisture or quality, or in kind of fuel.

Copes-Vulcan three-influence feedwater control has maintained the proper drum level while feeding exactly as needed for steam output. Regardless of load conditions, Copes-Vulcan control has maintained final steam temperature at the desired 760 degrees F.
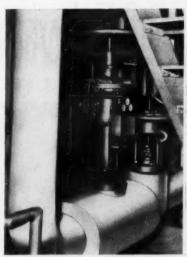
The complete story of this unusual installation is told in new Bulletin 1078. Write for it.

Central Control Panel. All recorders and controls for the bark-and-coal-fired boiler are centralized on this Copes-Vulcan panel.
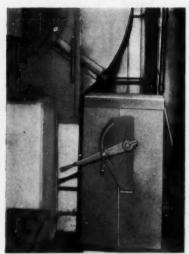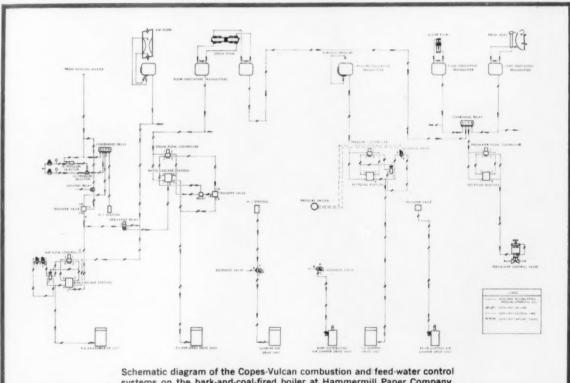
Spray Valve for Superheat Temperature Control. This Copes-Vulcan Type CV-D Spray Valve introduces cooling water into the superheater for dependably accurate steam-temperature control.

Feedwater-Flow Control Valve. This diaphragm-operated Type CV-D valve is part of the Copes-Vulcan three-influence feedwater control system

Drive Unit for Forced-Draft Fan Damper. This Copes-Vulcan drive unit, one of several in the control system, positions the forced-draft fan damper.

Schematic diagram of the Copes-Vulcan combustion and feed-water control systems on the bark-and-coal-fired boiler at Hammermill Paper Company.

Copes-Vulcan Division

BLAW-KNOX

and the Foolproof Check Valve

The design of the Edward tilting disk check valve started with a single objective: one-piece body construction. Edward designers were convinced that they could achieve this essential objective and, at the same time, assure ease of inspection and maintenance after the valve was welded in the line. Ideally, the valve also had to have these four important operating features: 1) Tight seating; 2) Non-stick operation; 3) Quiet, slamless closing; 4) Low pressure drop. Six years of research and testing gave them these desired features, and made possible this Edward success story.

ONE-PIECE CONSTRUCTION

Both manufacturing and engineering considerations indicated the wisdom of a design with the seat at a right angle to the pipe line. It was also essential that gravity close the disk, whether the pipe was in a horizontal or vertical position. Edward engineers solved this dual problem by welding a counterweight to the disk so that the force of gravity would pull the disk down tight on the seat whether the valve was installed horizontally or vertically. With this new disk idea, the valve seat could be formed of a single circular weld deposit of Stellite, and could be machined and finished through the end of the valve. The resultant one-piece body is virtually impervious to leakage or distortion.

TIGHT SEATING

Getting the disk to seat tightly was especially important because even a small amount of reverse flow was a hazard to pumps and compressors. Edward designers, after experimenting with several pivot positions, pivoted the disk half-way between its center and the upper seating surface. The tapered Stellite-faced disk closes on a tapered Stellite seat in the manner of a globe stop valve. When flow reverses, the larger lower portion of the disk swings downstream while the smaller upper portion swings upstream. Utilizing both gravity and the force of reverse flow, perfect alignment and tight seating are achieved.

SLAMLESS, NON-STICK OPERATION

The unique Edward water flow test loop with its 2,000,000 lb per hour capacity closely simulates actual service conditions. Edward researchers could observe the effects of vibration and water hammer at all stages of tilting disk development. Designing the valve to close quickly—before reverse velocity could build up—virtually eliminated slamming and vibration. Three features make this possible.

First, the disk is dome-shaped to avoid the hesitation of motion characteristic of flat disks. Second, disk travel is minimized by pivoting the disk in the flow stream. Finally, torsion springs enclosed in the pivot pins accelerate closing action from the full-open position. The valve closes efficiently and achieves maximum opening even at lowest flow rates.

LOW PRESSURE DROP

Minimum flow resistance was achieved through painstaking design of all surfaces in the flow line. Full port seat design plus streamlined disk and location of the counterweight out of the flow stream resulted in minimum pressure drop.

A FOOLPROOF VALVE

By tackling each problem separately—but always in relation to the entire design—Edward researchers came up with a tilting disk check valve that operated like no other check valve. It was dependable because it always functioned, effective because of its tight seating and low pressure drop, unique because it greatly minimized slamming, vibration, and water hammer.

This is the same tilting disk check valve, developed and patented by Edward Valves, that is so widely used in today's conventional, supercritical and nuclear power stations. Edward Catalog 14-Y gives full details.

Edward builds a complete line of forged and cast steel valves from ¼" to 24" for power, industrial, marine, and petroleum services. For more information, contact your Edward Representative, or write Edward Valves, Inc., 1206 West 145th Street, East Chicago, Indiana. Subsidiary of Rockwell Manufacturing Company. Represented in Canada by Lytle Engineering Specialties, Ltd., 438 St. Peter Street, Montreal. 6023
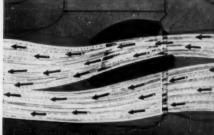
Director of Engineering and Research L. H. Carr and Plant Superintendent R. E. Rost observe test for disk position in relationship to pressure surge. Potentiometer indications of disk position are transmitted to and recorded by oscillograph in 1/100 second intervals.

Cut-away view of Edward tilting disk check valve shows torsion springs enclosed in pivot pins, pressure seal cover and other design features.

In the open position the disk rests against a stop, thus minimizing disk flutter. Careful design of disk facilitates maximum opening, even at extremely low flow rates.

How to feel good about a critical piping job

For the man who is responsible for the high-pressure, high-temperature piping in a power or processing plant, the wise decision is to delegate the fabricating and erecting *directly* to specialists. For instance, the experience and the modern shop & field facilities of Mitchell will make him happy from start to finish. Also, for built-in safety . . . ask us in.

W. K. MITCHELL & CO., INC.
Philadelphia 46, Pa.

WESTPORT JOINT
(PATENTED)

MITCHELL PIPING
SINCE 1899

PIPING FABRICATORS AND CONTRACTORS

16

Combustion

"Conference Maketh a Ready Man"

The very remarkable achievement of the second-manned Soviet space-ship, the Vostok II, which last week so captivated the world's imagination, drew from Nikita Khrushchev a public salute to the sizable team of technicians, scientists and engineers who made the feat possible. Some further contemplation on what all goes into such an achievement—the knowledge, the skill, the learning of all those who contributed—makes one realize the tremendous storehouse of basic science and practical know-how modern man has accumulated and presumably has at his command.

Francis Bacon some 400 years ago penned the observation "Reading maketh a full man, conference a ready man and writing an exact man." In the parlance of the street, "no truer words were ever spoken." And yet as Glenn Warren, then vice president of General Electric Co. and later to become president of ASME, so succinctly put it, "This brings us to another dilemma, namely, that the amount of these data—insufficient as I feel it is today—is still being put out in such profusion that the technical societies themselves are having difficulty handling it. Under these conditions the average scientist and engineer has great difficulty in finding and assimilating the available information and data even in his own field."

The solution, as Mr. Warren saw it and we are inclined to agree, was that some day this mass of data would be coded and keyed to a computer capable of printing a synopsis of the answers of all the pertinent reports on record and do the job in minutes. But until that day we must progress along a well worn avenue—the conference.

In all fields of endeavor the conference method has proved a highly valuable means of pooling thoughts, trying new ideas, achieving compromises, establishing a generally agreed upon course of action. In Bacon's words, making a man ready.

Within a month two vastly different but highly significant conferences of interest to the power industry are planned—the 4th Biennial Gas Dynamics Symposium, sponsored by the American Rocket Society and Northwestern University, with special emphasis on MHD, August 23–25; and the National Power Conference, sponsored by the ASME and the AIEE for San Francisco, Sept. 25–27. This latter meeting is one of the most comprehensive power programs if not the most comprehensive ever presented on the West Coast. It has always been our conviction that such conferences, properly planned and properly spaced, are vital assists to engineering progress. Attendance at these meetings, we think, will bear out our conviction.

Results indicate the vaporous carryover of sodium chloride is relatively low in the 2500–2600 psi pressure range so long as the salt concentration in the boiler water is maintained within reasonable limits. The magnitude of vaporous carryover obtained in the field tests reported herein wa appreciably less than that obtained by other investigators in laboratory model tests.   Possible explanations for this discrepancy are discussed.

By R. C. Ulmer[1] and H. A. Klein[2]

Combustion Engineering, Inc.

Impurities in Steam from High Pressure Boilers*

FROM the standpoint of the operator whose turbine is being fouled with deposits, it makes little difference whether the "Solids" present in his steam are the result of "Mechanical Carryover" or "Vaporous Carryover." The mechanism of "Solids" transport is extremely important, however, to the chemist or engineer who is trying to measure and control steam purity.

Before proceeding with further discussion, the quoted terms in the first paragraph should be defined. For the purposes of this paper, the word "Solids" will refer to all materials which are solid in the free-state at ambient temperatures. The term "Solids" may be applied even when such materials are in liquid solution or in the vapor phase. "Mechanical Carryover" refers to the presence of solids in the steam resulting from the entrainment of boiler water with the steam. "Vaporous or Volatile Carryover" will refer to the presence of solids in the steam due to volatilization of some component in the boiler water. "Total Carryover" will refer to the sum of the mechanical plus the vaporous carryover.

Mechanical carryover is a function of numerous factors including steaming rate, water-level, boiler design and the foaming characteristics of the boiler water. In general, mechanical carryover would be expected to be non-selective with respect to individual constituents since the entrained boiler water presumably contains the same ratio of constituents as the boiler water itself. Under conditions of pure mechanical carryover, the determination of a single constituent in the boiler water and steam will determine the per cent moisture content, and the ratio of all other components present.

Vaporous carryover, on the other hand, is completely selective in nature since it is dependent upon the volatility of a specific constituent under observation. Theoretically, for a given operating pressure, the ratio of the concentration of the vaporous component in the steam to its concentration in the boiler water is constant. This constant coefficient will be referred to as the distribution ratio. The ions present in the boiler water are not necessarily volatilized as such, and some intermediate product may be involved in the vaporization

process. Arbitrarily determined ratios may not be real distribution ratios, and caution must be employed in interpreting results.

Vaporous carryover is, however, independent of steaming rate, and thus may be readily differentiated from mechanical carryover. By reducing the steam velocity to a sufficiently low value, mechanical entrainment of boiler water may be completely eliminated. Any solids present in the steam under such conditions may be presumed to be due to vaporous carryover. This principle has been employed by many investigators to study the volatility of both silica and sodium salts.

Laboratory Tests by Other Investigators

All of the published data (1–5),[3] which could be found pertaining to the vaporous carryover of sodium salts, was based upon laboratory studies in model steam generators. The most comprehensive of these investigations was the work of Styrikovich (1). In this paper, it was theorized that the distribution ratios of all compounds are exponential functions of the density ratio of steam to water. A general equation, $K = (D_s/D_w)^N$, is presented where: K is the distribution coefficient, D_s/D_w is the ratio of the densities of steam and water, and N is the volatility constant for the specific compound. Coulter et al. (2) observed this same approximate relationship with regard to the distribution ratios for silica. Fig. 1 is a plot of the distribution coefficients of three sodium salts as a function of pressure based on the data of Styrikovich. On the basis of this plot, sodium hydroxide and sodium chloride volatilize to a significant degree at 2500 psi and above. Only at very high pressures, would the vaporous carryover of sodium sulfate be of any serious concern.

The distribution coefficients for sodium hydroxide and sodium chloride presented in Fig. 1 are sufficiently high to affect both the measurement and control of steam purity in high pressure boilers. If these values are presumed correct, the moisture content of high pressure steam can not be accurately determined by measurement of the conductivity or the sodium concentration of the steam when these salts[4] are present in the boiler water.

[3] Figures in parentheses refer to similarly numbered entries in Bibliography at the end of the article.
   [4] For uniform terminology, sodium hydroxide will be referred to as a salt.
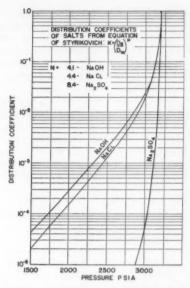
Fig. 1—The distribution coefficients for the steam-borne impurities charted above indicate if these salts are present they seriously affect accurate measurement of steam purity by conductivity devices

Furthermore, unless the concentration of these salts can be maintained within reasonable limits, their volatile carryover may result in turbine deposition.

Test Program—Purpose

To investigate these potential problems, the authors' Company instituted a test program to determine the magnitude of vaporous carryover in operating high pressure boilers. The purpose of these tests was to determine if the volatility effects predicted by laboratory investigators could be observed under field conditions.

There are numerous disadvantages to the performance of volatility studies in the field which are not present in the laboratory. A laboratory model boiler can be designed to generate steam at a sufficiently low rate to permit the assumption of "zero mechanical carryover." An expensive high-pressure boiler is rarely, if ever, operated at a sufficiently light load to insure this assumption. Furthermore, boiler water concentrations can be arbitrarily established in the laboratory to produce concentrations in the steam which can be analyzed by standard methods, and radioactive tracers may be effectively employed to greatly improve analytical precision. For obvious reasons, neither salts nor radioisotopes can be indiscriminantly added to a utility boiler. However, conditions in laboratory tests never really simulate conditions in the operating boiler. For this reason, the field test program was considered necessary to determine if the laboratory data could be applied to boiler operation.

Test Sites

The vaporous carryover test program was performed on three high pressure boilers belonging respectively to three different utilities. The first unit tested was the No. 4 boiler at the W. C. Beckjord Power Station of the Cincinnati Gas & Electric Co. At full load, this boiler produces about 1,200,000 lb per hr of steam. The second

test site was the No. 4 boiler at the Chesterfield Power Station of the Virginia Electric & Power Co. which produces 1,300,000 lb per hr of steam at full load. The third boiler tested was the No. 5 boiler of the Ashtabula Power Station of the Cleveland Electric Illuminating Co. The full-load steaming rate of this unit is 1,600,000 lb per hr. All three of these boilers are controlled-circulation units with operating drum pressures of from 2500 to 2600 psig. For brevity in the text, the three test sites may be referred to respectively as Beckjord, Chesterfield, and Ashtabula.

Although the boiler water specifications varied somewhat, the normal control in all three boilers may be described as a "low solids treatment" with the salt content in the boiler water generally maintained below 25 ppm. Slightly increased salt concentrations were permitted during the test periods to provide for more accurate analyses.

Test Procedure

Because of the relatively low solids content in the boiler water, sodium determinations were used as the basis for all carryover measurements. The instrument employed for analysis was a Beckman Model "B" flame spectrophotometer equipped to permit continuous sample analysis. Complete analyses were performed on most boiler water samples. Sodium was the only determination performed routinely on steam samples; however, a number of steam samples were concentrated to permit the analysis of other constituents and to check low-range sodium determinations.

Total carryover was determined by continuous sodium analysis of a flowing steam sample over a period of about an hour. Since total carryover must always be equal to or greater than the vaporous carryover, these measurements established maximum limit for vaporous carryover at the operating pressures of the respective boilers.

A special sampling system was designed to permit the determination of vaporous carryover, as such, at various pressures. A schematic of this system is shown in Fig. 2. Samples of steam and boiler water were simultaneously introduced into the pressure chamber at controlled rates. The mixture of steam and boiler water was then separated in the chamber with the water being discharged at the bottom and the steam at the top. The pressure in the chamber could be maintained at drum pressure by taking all of the pressure drop across the discharge valves, or
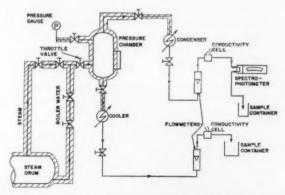
Fig. 2—Schematic of a special sampling system designed to permit determinations of vaporous carryover at various pressures

STEAM VELOCITY THROUGH THE PRESSURE CHAMBER AS A FUNCTION OF PRESSURE
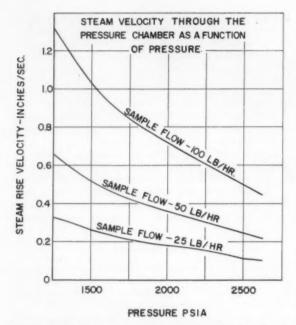
Fig. 3—Effectiveness of vapor separation is dependent upon a low steam-rise-velocity. The above chart shows the effect of pressure and sampling rate on this velocity

the pressure could be reduced to any desired lower pressure by taking a portion of the pressure drop across the inlet throttle valve. Effective control of pressure and flow was possible with the system as shown.

The effectiveness of separation is dependent upon the maintenance of a low "steam-rise-velocity" through the pressure chamber. This velocity is a function of both sampling rate and steam pressure. The relationship is illustrated in Fig. 3. No detectable mechanical carryover resulted within the pressure ranges tested with steam sample flows as high as 150 lb per hr. For convenience, however, most tests were performed with steam flows of from 25 to 50 lb per hr. In general, boiler water and steam flows were approximately equal.

Water level control in the pressure chamber was critical in the performance of these tests. A special method of water level control was integrated into the design of the pressure chamber. Fig. 4 is a sketch of the pressure chamber used in the tests performed at the Chesterfield and Ashtabula stations. Water level in the chamber was reflected by subcooling in the external water column. Thermocouple points which became submerged indicated temperature reductions of from 30–70 deg F. Thermocouple measurements were continuously monitored on a temperature recorder which permitted excellent control of water level. In the initial tests at Beckjord, a somewhat shorter pressure chamber with only three thermocouple points was used, and temperature measurements were made manually with a potentiometer. With this system, level control was not completely adequate and mechanical carryover was a problem on several occasions. For this reason, the chamber was redesigned for the subsequent tests. In general, water level was controlled between the first and second thermocouple points although mechanical carryover did not occur at normal steam flows unless the entire water column became submerged.

Considerable effort was taken to make the entire sampling system adiabatic. The inlet steam and boiler water lines were made as short as possible, and all surfaces from the boiler to the discharge valves were heavily insulated. Heat loss in the inlet lines is not particularly serious; however, precondensation will tend to dilute the boiler water in the pressure chamber. Condensation in the chamber itself is critical since post-condensation may establish a second water-steam equilibrium in the upper part of the vessel. This "washing" effect can reduce the apparent vaporous carryover. Every effort was made to prevent condensation in the pressure chamber by insulation. At Ashtabula the vessel was steam jacketed and then covered with insulation. The steam jacket was actually a tightly-wound coil supplied with steam at the same pressure as in the chamber. The possible effects of post-condensation on the test results will be discussed subsequently.

For the final tests which were performed at Ashtabula a small pressure-pot was installed in the sampling system to permit injection of salts directly into the pressure chamber. The purpose of this was to study the equilibrium between steam and concentrated boiler water. A concentrated salt solution was first injected into the steam-filled pressure chamber and then high pressure steam was introduced into the vessel below the water level and was discharged at the top. Once equilibrium was obtained, level was maintained constant. A small amount of condensate was introduced with the steam, and an equal amount of water was continuously discharged from the bottom of the vessel. This resulted in a gradual dilution of the boiler water but, at the same time, provided a sample for analysis. Both the steam and boiler water were continuously monitored over a wide range of concentrations. Concentrated solutions of sodium chloride and sodium sulfate were evaluated in this manner.

Test Results

Tables I and II present the total carryover test data obtained at the Chesterfield and Ashtabula stations. The data was obtained by continuous sodium analysis of the regular steam sample. Boiler water samples were intermittently collected for analysis. Tests of this type
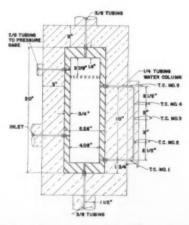
Fig. 4—Specially designed pressure chamber affords a better control of water level, a critical consideration in sampling vaporous carryover

TABLE I—TOTAL CARRYOVER TESTS

TABLE I—TOTAL CARRYOVER TESTS
Chesterfield Power Station—March 17, 1961
Drum pressure—2600 psig; load—170 megawatts

Steam Samples

Time	Sodium, ppm	Carry-over, %	Time	Sodium, ppm	Carry-over, %
4:46 p.m.	0.0027	0.036	5:04	0.0021	0.028
4:48	0.0027	0.036	5:05	0.0019	0.026
4:50	0.0023	0.031	5:07	0.0021	0.028
4:52	0.0021	0.028	5:08	0.0021	0.028
4:55	0.0018	0.024	5:09	0.0019	0.026
4:56	0.0021	0.028	5:10	0.0019	0.026
4:57	0.0021	0.028	5:12	0.0021	0.028
5:00	0.0018	0.024	5:15	0.0021	0.028
5:02	0.0021	0.028			

Boiler Water Samples

Time	Sodium, ppm	Chloride, ppm	Time	Sodium, ppm	Chloride, ppm
4:46	7.5	5.7	5:02	7.4	5.7
4:52	7.5	5.7	5:07	7.4	5.7
4:57	7.5	5.7	5:15	7.5	5.7

Average Sodium in Steam—0.0021 ppm
Average Sodium in Boiler Water—7.5 ppm
Average Total Carryover—0.028%

Average Sodium Chloride in Boiler Water—9.4 ppm
Maximum Possible Distribution Ratio for Sodium Chloride—
5.7×10^{-4}

TABLE II—TOTAL CARRYOVER TESTS
(Ashtabula Power Station—May 3, 1961)
Drum pressure—2500 psig

Steam Samples

Time	Sodium, ppm	Carry-over, %	Time	Sodium, ppm	Carry-over, %
1:02 p.m.	0.0034	0.032	1:11	0.0028	0.026
1:05	0.0026	0.025	1:13	0.0026	0.025
1:07	0.0030	0.028	1:15	0.0025	0.024
1:08	0.0035	0.033	1:17	0.0026	0.025
1:09	0.0030	0.028	1:18	0.0026	0.025
1:10	0.0032	0.030	1:25	0.0026	0.025

Boiler Water Samples

Time	Sodium, ppm	Chloride, ppm	Phosphate, ppm	Conductivity, micromhos	pH
1:00	10.6	9.3	1.1	64.2	10.0
1:15	10.6	9.3	1.0	66.0	10.0
1:20	10.6	9.1	1.1	68.5	10.0

Average Sodium in Steam—0.0029 ppm
Average Sodium in Boiler Water—10.6 ppm
Average Total Carryover—0.027%

Average Sodium Chloride in Boiler Water—15.3 ppm
Maximum Possible Distribution Ratio for Sodium Chloride—4.7
$\times 10^{-4}$

were not performed at Beckjord; however, spot measurements of total carryover indicated the same approximate relationship.

It may be noted that some sodium values have been reported to nearest tenth of a part-per-billion. The authors do not presume the test to be accurate to this degree; however, it was believed desirable to report the data as closely as the calibration curve could be read.

Because the sodium concentrations in the steam were always in the low "part-per-billion range," methods for concentrating steam samples were occasionally employed to confirm the test results. One of the methods employed was that proposed by W. B. Gurney (6). In this procedure, the high pressure steam sample was throttled to a lower pressure which condensed a percentage of the steam. This condensate absorbed virtually all of the solids originally present in the steam. By separating this moisture from the steam in the pressure chamber, the solids present in the steam could be concentrated by some factor dependent upon the percentage of moisture formed. The advantage of this technique was that it was quite rapid and permitted direct flow analysis. A second concentration procedure employed a "continuous sample evaporator." This equipment permitted the evaporation (without boiling) of a large steam sample to a relatively small volume which could then be analyzed as desired. With this technique, virtually any number of concentrations could be obtained; however, the process was more time consuming than the "expansion" procedure. Both concentration methods showed good

agreement with the direct sodium analyses. Some representative values are presented in Table III.

Of primary interest in the test program were the direct determinations of vaporous carryover under typical boiler water conditions. However, to insure that the boiler water would contain one of the two salts predicted to have a significant volatility, the stations involved were requested to introduce a small amount of sodium chloride into the boilers to be tested. Sodium chloride was selected over the sodium hydroxide because of possible side effects associated with the latter. Each of the three utilities consented to the request, and from 10 to 25 ppm of sodium chloride were maintained in the water in these boilers during the test period. Several test runs were performed before the addition of sodium chloride to the boilers for comparative purposes.

Each test run consisted of measurements with the chamber at full drum pressure and at a number of reduced pressures. The test range included roughly from 1500 to 2600 psig. Table IV presents a resume of all the vaporous carryover data collected at the three test sites for easy comparison.

Only at Ashtabula was the sampling system equipped to permit evaluation of vaporous carryover from highly concentrated boiler water solutions. Sodium chloride and sodium sulfate were injected into the pressure chamber on separate test runs. After equilibrium was obtained, the steam and boiler water samples were continuously monitored over a wide range of concentrations. Tables V and VI present a resume of the data obtained.

TABLE III—TECHNIQUES FOR CONCENTRATING STEAM SAMPLES

Station	Pressure, psi	Steam Sample Direct Sodium Measurement, ppm	Method of Concentration	No. of Concentrations	Sodium in Concentrate, ppm	Boiler Water Sample Sodium, ppm	Phosphate, ppm	Chloride, ppm
Beckjord	2550	0.0010	Expansion	10	0.0130	2.6	. . .	. . .
Beckjord	2550	0.0010	Expansion	5	0.0055	2.6	. . .	. . .
Chesterfield	2575	0.0014	Evaporation	27.2	0.0392	8.7	1.2	6.6
Chesterfield	2375	0.0013	Evaporation	35.2	0.0440	8.4	1.2	6.6
Ashtabula	2450	0.0021	Evaporation	34.2	0.0730	8.0	3.0	6.6
Ashtabula	2550	0.0018	Evaporation	43.1	0.0710	9.4	2.2	7.7

Discussion of Results

The results shown in Tables I and II indicate average total carryover values of 0.028 per cent and 0.027 per cent for boilers operating at pressures of 2600 psig and 2550 psig, respectively. The "limiting" distribution ratios for sodium chloride were calculated by presuming that all of the carryover was vaporous and that sodium chloride was the only volatile salt present. The values $(5.7 \times 10^{-4}, 4.7 \times 10^{-4})$ obtained were considerably less than those predicted using the equation of Styrikovich (see Fig. 1).

The direct determinations of vaporous carryover (Table IV) produced even lower distribution ratios than those shown above. The sodium concentrations in the steam samples were all extremely low, and all distribution coefficients obtained were less than those predicted by other investigators. The possibility of post-condensation in the upper part of the pressure chamber was immediately suspected. Even a small amount of condensation can absorb a large percentage of the "vaporized solids" from steam when the distribution ratio is small. If a second water-steam equilibrium occurs in the upper part

TABLE IV—VAPOROUS CARRYOVER TESTS

W. C. Beckjord Power Station—February 25-28, 1961

Pressure, psig		Sodium, ppm	Chloride, ppm	Phosphate, ppm	Conductivity, mmho	$K = \dfrac{Na(Steam)}{Na(B.W.)}$
2500	Steam	0.0010	...	...	2.80	1.4×10^{-4}
	Water	7.2	9.2	...	35.0	
2000	Steam	0.0007	...	...	2.75	6.9×10^{-5}
	Water	10.2	13.1	...	49.6	
1500	Steam	0.0006	...	...	2.80	5.2×10^{-5}
	Water	11.6	16.2	...	57.0	
2500	Steam	0.0020	...	...	1.45	2.0×10^{-4}
	Water	10.0	13.1	...	41.6	
2500	Steam	0.0008	...	...	2.02	7.1×10^{-5}
	Water	11.2	13.8	...	41.0	
1950	Steam	0.0007	...	...	2.10	6.3×10^{-5}
	Water	11.2	13.5	...	38.9	
1450	Steam	0.0005	...	...	2.00	2.8×10^{-5}
	Water	18.0	21.50	...	86.5	

Chesterfield Power Station—March 14-16, 1961

Pressure, psig		Sodium, ppm	Chloride, ppm	Phosphate, ppm	Conductivity, mmho	$K = \dfrac{Na(Steam)}{Na(B.W.)}$
2600	Steam	0.0013	...	...	2.54	2.5×10^{-4}
	Water	5.2	...	2.2	35.1	
2400	Steam	0.0009	...	...	2.45	1.8×10^{-4}
	Water	5.0	...	2.1	36.0	
2200	Steam	0.0007	...	...	2.60	1.4×10^{-4}
	Water	5.2	...	2.0	40.4	
2025	Steam	0.0005	...	...	2.46	9.6×10^{-5}
	Water	5.2	...	2.0	37.0	
1800	Steam	0.0003	...	...	2.35	5.4×10^{-5}
	Water	5.6	...	2.0	35.4	
1600	Steam	0.0003	...	...	2.55	5.7×10^{-5}
	Water	5.3	...	2.0	36.9	
2475	Steam	0.0010	...	...	2.50	1.3×10^{-4}
	Water	7.6	...	6.5	52.8	
2175	Steam	0.0009	...	...	2.07	1.2×10^{-4}
	Water	7.8	...	6.0	59.8	
1975	Steam	0.0007	...	...	2.25	9.0×10^{-5}
	Water	7.8	...	6.0	60.2	
1800	Steam	0.0007	...	...	2.26	9.0×10^{-5}
	Water	7.8	...	6.2	59.8	
1600	Steam	0.0006	...	...	2.32	8.0×10^{-5}
	Water	7.5	...	6.35	58.5	
2575	Steam	0.0014	...	...	1.86	1.6×10^{-4}
	Water	8.7	6.6	1.23	51.8	
2375	Steam	0.0013	...	...	2.14	1.5×10^{-4}
	Water	8.4	6.6	1.23	51.9	
2225	Steam	0.0011	...	...	2.15	1.4×10^{-4}
	Water	8.0	6.1	1.75	56.0	
2000	Steam	0.0010	...	...	2.17	1.3×10^{-4}
	Water	8.0	6.3	1.24	55.5	
1800	Steam	0.0009	...	...	2.25	1.1×10^{-4}
	Water	8.2	6.6	1.24	55.6	

Ashtabula Power Station—April 30–May 1, 1961

Pressure, psig		Sodium, ppm	Chloride, ppm	Phosphate, ppm	Conductivity, mmho	$K = \dfrac{Na(Steam)}{Na(B.W.)}$
2450	Steam	0.0021	...	...	2.14	2.6×10^{-4}
	Water	8.0	6.6	3.0	57.5	
2200	Steam	0.0019	...	...	2.10	1.9×10^{-4}
	Water	10.0	8.3	3.2	69.0	
1900	Steam	0.0019	...	...	2.20	1.9×10^{-4}
	Water	9.8	8.3	3.4	67.0	
1750	Steam	0.0015	...	...	2.15	1.6×10^{-4}
	Water	9.4	7.7	3.2	63.5	
1500	Steam	0.0015	...	...	2.20	1.5×10^{-4}
	Water	9.8	7.9	3.2	65.5	
2250	Steam	0.0019	...	...	1.68	2.0×10^{-4}
	Water	9.4	7.7	2.2	66.6	
2050	Steam	0.0012	...	...	1.67	1.4×10^{-4}
	Water	8.3	7.0	2.0	62.5	
1750	Steam	0.0015	...	...	1.67	1.9×10^{-4}
	Water	7.8	6.6	2.4	57.2	
1400	Steam	0.0010	...	...	1.64	1.3×10^{-4}
	Water	7.5	6.6	2.4	56.4	

TABLE V—SODIUM CHLORIDE INJECTION TEST

Ashtabula Power Station—Unit No. 5

Cleveland Electric & Illuminating Co.—May 2, 1961

Pressure—2550 psig

	Boiler Water		Steam		
Time	Conductivity in Micromhos	Sodium PPM	Conductivity in Micromhos	Sodium PPM	K × 10⁻⁴
6:51	6600	1590	...	...	...
6:53	5550	1330	...	...	...
6:54	5040	1290	...	...	...
6:55	4850	1120	7.11	0.800	7.14
7:04	2100	400	5.9	0.400	10.0
7:05	2000	380	5.11	0.300	7.89
7:06	1780	340	4.71	...	...
7:07	1650	310	2.74	0.175	5.65
7:08	1540	290	2.66	0.140	4.83
7:09	1450	270	2.51	0.125	4.63
7:10	1320	245	2.45	0.106	4.32
7:11	1130	210	2.9	0.096	4.57
7:13	935	183	2.81	0.088	4.81
7:14	836	163	2.25	0.075	4.60
7:15	770	150	2.16	0.060	4.00
7:16	701	138	2.1	0.047	3.41
7:17	638	125	2.06	0.037	2.96
7:18	583	113	2.04	0.031	2.74
7:19	540	105	2.02	0.027	2.57
7:20	496	97	1.99	0.022	2.27
7:21	452	87	1.99	0.021	2.41
7:22	405	79	1.95	0.022	2.78
7:23	353	69	1.95	0.019	2.75
7:24	322	62	1.95	0.014	2.26
7:25	300	58	1.94	0.017	2.93
7:26	275	55	1.91	0.014	2.55
7:27	264	53	1.91	0.011	2.08
7:28	242	48	1.89	0.0089	1.85
7:29	224	43	1.88	0.0065	1.51
7:30	205	39	1.88	0.0060	1.54
7:31	192	37	1.88	0.0050	1.35
7:32	169	33	1.89	0.0040	1.21
7:33	155	30	1.90	0.0035	1.17
7:34	140	26	1.90	0.0030	1.15
7:35	126	24	1.89	0.0030	1.25
7:36	117	22	1.86	0.0030	1.36
7:37	98.5	20	1.85	0.0025	1.25
7:38	90.3	18	1.86	0.0025	1.39
7:39	84.4	15	1.86	0.0023	1.53
7:40	78.0	14	1.86	0.0023	1.64
7:41	68.6	12	1.85	0.0023	1.91
7:42	61.5	12	1.86	0.0021	1.75
7:43	54.9	11	1.86	0.0020	1.82
7:44	49.2	10	1.86	0.0020	2.00
7:45	44.6	9	1.85	0.0018	2.00
7:46	39.7	8	1.84	0.0016	2.00
7:47	34.4	6	1.84	0.0015	2.50

Fig. 5—The possibility of a post condensation occurring in the test vessel will introduce a serious error in sampling. The above plot gives the maximum possible error from this post condensation

the effective "washing" which could occur. A more convincing factor was the fact that the total carryover test results were of the same order of magnitude as the vaporous carryover data. Total carryover tests were performed on completely condensed samples taken directly from the boiler. Condensation could have little effect on these determinations. Nevertheless, it was decided that the tests to be performed at Ashtabula would incorporate "steam-jacketing" in addition to the normal insulation on the pressure chamber. The results obtained with steam-jacketed equipment approximated those obtained at Beckjord and Chesterfield.

TABLE VI—SODIUM SULFATE INJECTION TEST

Ashtabula Power Station—Unit No. 5

Cleveland Electric & Illuminating Co.—May 2, 1961

Pressure-2550 psig

	Boiler Water		Steam		
Time	Conductivity in Micromhos	Sodium PPM	Conductivity in Micromhos	Sodium PPM	K × 10⁻⁵
5:53	3400	810	...	0.058	7.16
5:54	3350	790	...	0.062	7.84
5:55	3400	810	...	0.060	7.41
5:56	3300	780	...	0.056	7.18
5:58	3200	760	...	0.056	7.37
5:59	3100	720	...	0.060	8.33
6:00	3000	700	...	0.053	7.57
6:02	2200	480	...	0.046	9.58
6:03	2100	460	...	0.043	9.35
6:04	1900	410	...	0.034	8.29
6:05	1800	380	...	0.034	8.95
6:06	1700	360	...	0.033	9.17
6:07	1600	340	...	0.034	10.00
6:08	1500	310	...	0.037	11.93
6:09	1450	300	...	0.034	11.33
6:10	1400	290	...	0.031	10.69
6:11	1300	265	...	0.030	11.32
6:12	1200	240	...	0.027	11.25
6:13	1150	215	...	0.025	11.63
6:14	1035	200	...	0.023	11.50
6:15	930	195	...	0.023	11.79
6:16	890	190	...	0.021	11.05
6:17	830	182	...	0.017	9.34
6:18	755	164	...	0.013	7.93
6:19	680	148	...	0.0125	8.45
6:20	615	132	...	0.0115	8.71
6:21	580	125	...	0.0100	8.00
6:22	550	120	...	0.0092	7.67
6:23	512	110	...	0.0077	7.00
6:24	500	108	...	0.0065	6.02

of the test vessel, the steam in contact with the condensate would be effectively "washed." As a consequence, the "apparent" distribution ratio would be lower than the actual equilibrium value. The discrepancy between the two values is a function of the percentage condensation, the actual distribution ratio, and the effective contact between the steam and condensate. Fig. 5 is an illustration of the potential error which can result from "post-condensation" in a test vessel. This plot assumes that all condensate is retained in the test vessel and that the condensate absorbs solids to saturation. The plot, therefore, gives the maximum possible error which can be introduced by the "washing" phenomena.

Although heat losses can never be completely prevented by insulation alone, post-condensation was not believed to have been a problem at Beckjord or Chesterfield. The reason for this opinion was that the results were not affected by significant changes in steam flow through the pressure chamber. Presuming a constant rate of heat loss, an increased steam flow will reduce the percentage of condensate formed and should thus reduce
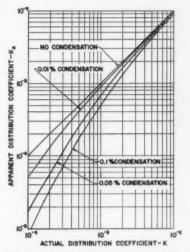
The salt injection tests were conceived in an effort to obtain some vaporous carryover data under simulated conditions of high boiler water concentrations. This was done by first injecting a concentrated salt solution into the pressure chamber and then continuously passing high pressure steam through the chamber and out the top. The steam was introduced below the water level to promote mixing and contact. These conditions are not identical with those in a model steam generator since the steam does not originate from the concentrated water; however, if contact between the water and steam is adequate, similar results might be expected.

The data obtained in the sodium chloride injection tests are rather interesting. The calculated distribution ratios are definitely higher with high salt concentrations in the water. This phenomenon may be peculiar to the type of test performed; however, it is noteworthy that the values obtained at the high concentrations are in fair agreement with the laboratory data of other investigators while the values obtained at low concentrations are in agreement with the coefficients we had previously obtained.

It is not unreasonable that a measured distribution coefficient might vary with concentration. This would depend upon the mechanism by which the volatile component was formed. For the sake of discussion, let us assume that sodium chloride is vaporized as a result of the instantaneous formation of a "neutral sodium chloride molecule" which is volatile under high pressure boiler conditions:

$$Na^+ + Cl^- \xrightarrow{k_1} NaCl^{\circ}{}_{(w)} \xrightarrow{k_2} NaCL_{(s)} \quad (1)$$

$$K_1 = \frac{[NaCl^{\circ}]_w}{[Na^+][Cl^-]} \quad (2)$$

$$K_2 = \frac{[NaCl]_s}{[NaCl]_w} \quad (3)$$

Since the formation of the neutral molecule is dependent upon the reaction of sodium ions with chloride ions, its concentration is proportional to the sodium ion concentration times the chloride ion concentration. In a stoichiometric solution, therefore, the concentration of the volatile component is proportional to the square of the ionized salt concentration:

$$[Na^+.Cl^-] = [Na^+] = [Cl^-]$$
$$\text{Stoichiometric solution} \quad (4)$$

$$K_1 = \frac{[NaCl^{\circ}]}{[Na^+][Cl^-]} = \frac{[NaCl^{\circ}]}{[Na^+.Cl^-]^2} \quad (5)$$

Thus, even though the actual distribution ratio is presumed constant (Equation 3), the volatile sodium chloride concentration in the steam would be proportional to the square of the ionized salt concentration in the boiler water. Regardless of the analytical technique employed, the measured distribution ratio would be a function of the boiler water concentration and not a constant.

The authors do not wish to imply that Equation (4) is the mechanism by which sodium chloride is volatilized. The data collected thus far are not sufficient to justify such a conclusion. However, it is possible that an intermediate product is involved in the vaporization process. Volatilization of silica from alkaline boiler water is gen-
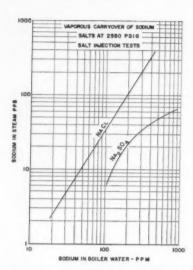
Fig. 6—Comparison between the distribution coefficients for two sodium salts is afforded from the above plot

erally attributed to the formation of silicic acid by hydrolysis. The dependency of vaporization upon this intermediate product causes the carryover of silica to be a function of the pH of the boiler water.

The distribution coefficients obtained for sodium sulfate (Table VI) in the salt injection tests varied somewhat with concentration; however, they do not show the same pattern as do the results for sodium chloride. For comparison purposes, data from both tests are plotted together in Fig. 6. The vaporous carryover of sodium sulfate was always much lower than that for sodium chloride. This, of course, was expected.

The lack of agreement between our field data and the results obtained by others in the laboratory was of concern to the authors throughout the test program. Several possible explanations were considered. The possible influence of concentration on distribution ratios is significant since there is a question as to whether adequate laboratory tests have been performed at the low concentrations which are typical in high pressure boilers. Although both Styrikovich (1) and Sastry (3) indicate that their tests were performed over a range of concentrations, no actual data was presented in the copies of the translations which were available. Neither Straub (4) nor Morey (5) performed volatility tests in the low concentration range.

The steam-water dynamics in a boiler were compared with those in a simulated test vessel to evaluate any fundamental differences. Introduction of feedwater into a boiler results in subcooling in the drum. The degree of subcooling depends upon the temperature of the feedwater, the circulation ratio, and the extent of mixing between the feedwater and the boiler water. In a large high pressure unit, the mean temperature of the water in the drum might be as much as 25 deg F below saturation. The temperature of the water in contact with the steam phase would, of course, be somewhat higher than the average. It is difficult to predict what influence this factor might have on vaporous carryover. The water in contact with the steam during generation and separation is obviously at saturation temperature, and the degree of contact between the steam and the water

in the drum is questionable. However, a few degrees of temperature could have a significant effect on salt volatility, and the vaporous carryover might be less in an operating boiler than in a model steam generator where the boiler water is maintained at saturation temperature.

Strictly on the basis of the data obtained in the field, vaporous carryover does not appear to present a problem in the 2500-2600 psi pressure range provided that reasonable limits are placed on boiler water concentrations. In the boilers tested, the total carryover of sodium is relatively low even with higher-than-normal concentrations in the boiler water. Data collected under these conditions indicated a distribution ratio for sodium chloride which was definitely less than 5×10^{-4} and probably less than 2×10^{-4}. Converted to a more familiar form, these figures represent 0.05 per cent and 0.02 per cent carryover, respectively. Although primary emphasis was placed in the study of sodium chloride characteristics, sodium phosphate and sodium hydroxide were present in many of the boiler water samples tested. Variations in these constituents did not appear to affect the overall pattern of low vaporous carryover. It is difficult to extrapolate the data obtained to predict conditions in boilers operating at higher pressures. If distribution ratios are exponential functions as many investigators believe, the magnitude of vaporous carryover could increase rapidly with only a moderate increase in pressure.

Vaporous carryover can be expected to complicate the determination of steam purity in high pressure boilers. If volatile carryover is significant, it will be virtually impossible to accurately measure the moisture content of the steam by chemical means. Where sodium analysis is employed to determine solids content, considerable question may exist as to what anions, if any, are also present in the steam. Vaporous carryover may also present a problem in steam sampling. A small amount of condensation may absorb a large percentage of the vaporized solids in a steam sample. The sample line should be designed to insure that any condensate formed can not drain back into the drum. The total sample leaving the drum should be collected for analysis. None should be by-passed; else, the sample collected might contain all or none of the concentrated condensate.

The data presented in this paper represents only a portion of the test work which the authors' Company intends to perform on the study of "Impurities in Steam from High Pressure Boilers." It is anticipated that future work will include studies of the characteristics of sodium hydroxide, sodium phosphate, and sodium sulfate.

Summary and Conclusions

A research program was undertaken to determine the extent to which sodium salts are subject to vaporous carryover under high pressure boiler conditions. Field tests were performed on three boilers operating in the 2500-2600 psi pressure range. Primary emphasis was placed in the study of sodium chloride although other salts were evaluated to a lesser degree.

The results obtained indicate that the volatility of sodium chloride is relatively low at these pressures so long as the salt concentration in the boiler water is maintained within reasonable limits. With from 10 to 25 ppm of sodium chloride in the boiler water, calculated distribution ratios averaged less than 2×10^{-4}, or 0.02 per cent carryover. Under these conditions, vaporous carryover would result in the presence of approximately one part-per-billion of sodium in the steam. On this basis, vaporous carryover of sodium salts does not appear to present a serious problem with regard to turbine deposition in the 2500-2600 psi pressure range.

The magnitude of the vaporous carryover observed in the field tests was considerably less than that obtained by other investigators in the laboratory. There are several possible explanations for this lack of agreement; however, present information on the mechanics of vaporous carryover is inadequate to permit evaluation of all potential factors.

The authors' Company plans further research work with regard to carryover in high pressure boilers. Future tests will include studies of the characteristics of sodium hydroxide, sodium sulfate, and sodium phosphate.

Acknowledgment

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Computer Control System for 1300 MW TVA Plant

The Tennessee Valley Authority will use two digital computers to control its Paradise Station Units 1 and 2 near Drakesboro, Kentucky. The computers, supplied by Thompson Ramo Wooldridge Inc., will be used in conjunction with combustion control equipment being supplied by Republic Flow Meters Co.

The system will automatically control the turbine generators, sequence monitor the steam generators, and will sequence cold start, hot restart, normal shutdown, or emergency shutdown of the units. In addition, the system will perform sensor scanning and alarm monitoring, data logging, trend recording and performance calculations.

The two units will produce a total of 1,300,000 kilowatts of power, and will be the largest power generating installations in the United States. The first unit is scheduled to begin operation in the fall of 1962; the second is scheduled for mid-1963.

Unlike conventional electrical or pneumatic valves, new electro-hydraulically operated Hydramotors remain in last position when power is removed — until energized to trip. Designed by General Controls for use on the main gas and oil fuel lines of power plants, these Hydramotors are especially built and internally wired to close tightly in just one second — *but only on operator command.*

This eliminates the possibility of shutdown in case of power source failure...keeps flow to boilers constant at all times because these valves are electrically operated. No air pressure or pneumatic piping is required.

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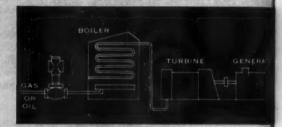
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An auxiliary switch unit, with three switches that are independent of Hydramotor controls, is available to interlock the shutoff valve with other power plant systems. Each switch is adjustable to trip at any position of valve stroke. This explosion proof, weather proof unit can be mounted in the field. Six SPDT switches may be obtained by using two of the units with a Hydramotor.

Write today for specifications and complete information. Ask for Bulletin SD1-H2, 3KA, C-1 (Hydramotor Shutoff Valves) and Bulletin 608.287 (Auxiliary Switch Unit).
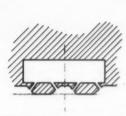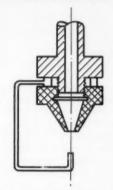
Fig. 7—Photos illustrate the spray formation for the jet nozzle types illustrated and identified in the sectional views immediately below each photo

IMPINGING JET NOZZLE

SOLID SURFACE IMPACT NOZZLE

SWIRL NOZZLE

Atomization of Liquid Fuel for Combustion — II*

By PAUL EISENKLAM†

Imperial College of Science and Technology

Atomization in a liquid fuel combustion system is considered as a mass transfer operation, and the factors influencing the rate of evaporation of drops and sprays are discussed. The mechanisms of liquid disintegration are detailed and the practical means of achieving clouds of finely divided drops from atomizers are described.

Atomizers can be conveniently grouped according to the primary source of energy which is employed to achieve disintegration. Thus, one distinguises *pressure atomizers* in which pressure energy is employed, *rotary atomizers* in which one part of the atomizer rotates thereby imparting centrifugal energy to the liquid, and *twin fluid* or *blast atomizers* in which a gas is used which impinges on the liquid so as to bring about disintegration. In Table 3, p. 29, COMBUSTION, July 1961, the conventional atomizers are listed and classified. (Combinations of types are shown by arrows.)

A great variety of designs of atomizers is used in combustion appliances; the reader is referred to other publications for descriptions or methods of control, such as those given in references (23) and (24).

PRESSURE ATOMIZERS

Pressure atomizers produce fan or conical sprays. The former are obtained by impact either of two jets as in the
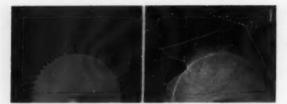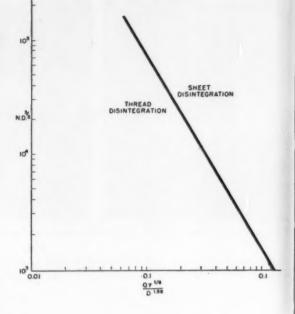
Fig. 8—Liquid disintegration from a spinning disk

Liquid: heavy fuel oil. $\rho = 0.89 \ g/cm^3$
$\gamma = 21 \ dynes/cm; \ N = rpm.; \ D = cm$
$Q = cm^3/sec; \ \eta = poise$

Fig. 9—Type of liquid disintegration for spinning cup atomizers under given operating conditions of a heavy fuel of above characteristics
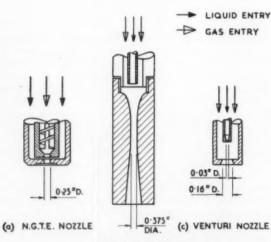
impinging jet nozzles, or by a jet on a solid surface as in the solid surface impact nozzles of the deflector nozzles, where the jet is not freely formed but is deflected at the orifice. By suitably shaping the inlet and outlet portions of an orifice, impact can be obtained inside the nozzle as in fan spray nozzles. Conical sprays are obtained from swirl spray nozzles by inducing a swirling motion in a swirl chamber upstream of the orifice. Where an air core is formed in the chamber, the spray will be hollow resulting in an appreciable spray angle. Where this air core is suppressed, a narrow spray or drowned spray results. Divergent pintle nozzles are deflector nozzles where the liquid impacts on a suitably shaped pintle in the centre of an orifice resulting in radial flow in a conical sheet. Fig. 7 shows the spray formation close to the atomizer

for a plug-type swirl nozzle, an impinging jet nozzle and a solid surface impact nozzle.
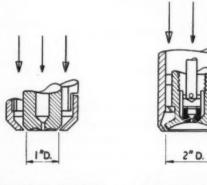
ROTARY ATOMIZERS

Rotary atomizers invariably consist of a spinning cup into which the liquid is suitably introduced at the inside near the axis resulting in a liquid sheet flowing inside the cup and leaving it at its periphery. It is essential, if fine atomization is to be achieved, that the operating conditions are such that the liquid at the rim is in the form of a sheet and not in the form of threads. (The conditions of operation for the formation of drops at the rim are outside the practical limits under which industrial combustion appliances operate.) In Fig. 8 these two forms of disintegration are shown from the rim of a flat

(a) N.G.T.E. NOZZLE

(b) B.I.S.R.A. NOZZLE

(c) VENTURI NOZZLE

INTERNAL MIXERS

(d) SIMPLE VORTEX CUP
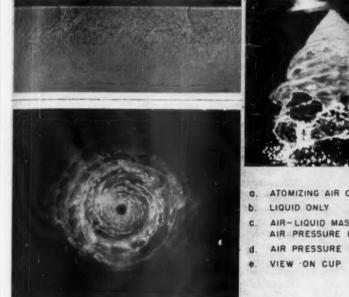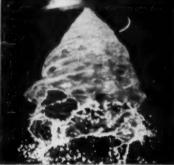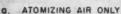
(e) VORTEX CUP WITH SWIRL SPRAY NOZZLE

EXTERNAL MIXERS

Fig. 10—Some twin fluid atomizers

a. ATOMIZING AIR ONLY
b. LIQUID ONLY
c. AIR–LIQUID MASS RATIO 0.75
   AIR PRESSURE 10 P.S.I.
d. AIR PRESSURE 15 P.S.I.
e. VIEW ON CUP

Fig. 11—Development of a spray from a vortex cup atomizer

spinning disk. The type of disintegration which can be expected from the operating conditions can be predicted from the correlation shown in Fig. 9.

Twin Fluid Atomizers

Twin fluid atomizers make use of low-, medium- or high-pressure air which is introduced to the atomizer in such a way that it impinges either on a liquid sheet formed by any of the devices already described or simply on a liquid jet. Where the impingement and subsequent mixing of the two fluids occurs inside the atomizer, the atomizer is called an internal mixer. In an external mixer no mixing of the two fluids occurs inside the atomizer so that the flow of each can be controlled independently. Only the twin fluid atomizers referred to later under that heading are shown in Fig. 10. Of the three internal mixers, the venturi-type atomizer is one of the oldest designs and has found wide application in steel furnace in the form of the BISRA nozzle. The liquid is injected through a pipe upstream of the entrance to a conventional convergent or convergent-divergent gas nozzle. In the NGTE nozzle, a thin sheet of liquid is made to flow radially inwards and is disintegrated by a centrally placed swirling air jet. The two external mixers are both of the vortex type. Liquid issues from a central orifice or through a swirl spray nozzle and moves on the inside of a cup at the periphery of which it meets a sheet of gas at right angles. The flow external to the atomizer is such that vortices are formed close to the cup which constrain the injected liquid to move along the cup. The flow from an atomizer of type Fig. 10d is shown in Fig. 11e and that of Fig. 10e in Fig. 11b, c and d.

The Performance of Atomizers

There are three major criteria by which the performance of an atomizer is assessed, viz., the sizes and op-

erating conditions required to give a desired liquid flow-rate, the spatial configuration of the resulting spray and its mean drop size and size dispersion.

Flow-rate

For pressure atomizers, the applied pressure head is completely converted into velocity head and over a wide range of pressures, the flow-rate is proportional to the square root of the pressure. The proportionality constant has been termed "Flow-number" and is usually given in dimensionally inconsistent units, viz.

$$\text{Flow-number (FN)} = \frac{\text{flow-rate (gallons per hour)}}{\sqrt{\dfrac{\text{pressure differential}}{\text{across orifice (psi)}}}} \quad (11)$$

In practice, the flow-rate varies more nearly to the 0.4th power of the pressure than to the 0.5th but the flow-number, as defined above, is fairly constant for each size of orifice and liquid, viz.

$$FN = 293 \frac{C_Q \cdot A}{\sqrt{\rho_l}} \quad (12)$$

where

A = area of orifice (cm^2)
C_Q = discharge coefficient
ρ_l = density of liquid (g/cm^3)

For fan spray nozzles, the value of C_Q is between 0.8 to 0.95 and is independent of the viscosity of the liquid.

For swirl spray nozzles, C_Q varies with Reynolds number (based on the orifice diameter) and is of the order of 0.2 to 0.4. It is not independent of the viscosity of the liquid because of frictional effects. If the effect of viscosity on the tangential velocity is predominant,
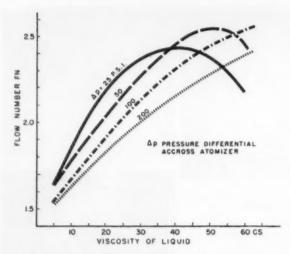
Fig. 12—Variation of flow-number with pressure and viscosity

a decrease in viscosity will bring about a decrease in discharge coefficient because of the increased frictional drag brought about by an increased velocity. If, on the other hand, the axial velocity is predominantly affected, a decrease in viscosity will bring about an increase in discharge coefficient because of a decrease in the air core diameter. The effect of viscosity on discharge coefficient is therefore complex and can only be assessed empirically. In Fig. 12 (reconstructed from reference (25), Fig. iv-5 is shown the change of discharge coefficient (expressed as a flow-number) with viscosity and pressure differential for a swirl nozzle of the *monach*-type spraying oil at room temperature.

For twin fluid atomizers of the external mixing type similar considerations apply since the liquid is supplied
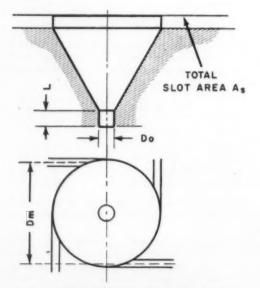

Fig. 13—Sketch illustrating the dimensions of a swirl nozzle

through an orifice. There are, of course, limiting flow-rates to be avoided below which no sheet is formed in any atomizer. Under these conditions, the pressure and twin fluid atomizers will produce a liquid jet, and the rotary cups will not be evenly wetted so that the liquid will leave the rim in an irregular thread pattern.

THE SPATIAL CONFIGURATION OF THE SPRAY

This criterion includes not only the geometrical extent of the spray but also the spatial dispersion of the drops.

Most atomizers produce an angular spray, either a triangular sheet of a particular spray angle or a cone of corresponding cone angle. These angles are affected by the surrounding atmosphere and the applied pressure differential and one distinguishes between theoretical angles which are independent of these variables relating to sprays in a quiescent atmosphere and practical angles under operating conditions. Theoretical angles for fan spray nozzles can at present not be predicted simply from their designs, but nozzles are made which give angles within the range of 30 deg to 180 deg. For swirl spray nozzles the cone angle is the inverse tan of the ratio of the tangential to the axial velocity at the exit and it can be predicted from the design of the swirl chamber. The parameter expressing the dimensions of the nozzle (as shown in Fig. 13) is:

$$\frac{A_s}{D_0 D_m} \left(\frac{D_m}{D_0}\right)^{1-n}$$

where

A_s = cross-sectional area of swirl ports
L = length of orifice
D_0 = orifice diameter
D_m = average diameter of swirl chamber at entry to swirl ports
n = an index

The index n is smaller than unity, viz., between 0.1 and 0.5. Its value depends not only on the type of nozzle and the surface finish of the passages but also on the viscosity of the fluid. This dependence is of a complex nature because of the reasons stated above. Some experimental results over a wide range of sizes are shown in Fig. 14. They have been obtained with water at differential pressures from 50 to 200 p.s.i. The discharge coefficient is related to the cone angle; both are plotted in Fig. 14 against the same parameter. Practical angles vary with atmospheric conditions and pressure because of the pressure differentials which are set up between the inside of the spray-cone and the surrounding results in entrainment. Fig. 15, (26), shows variation of the practical spray angle for a swirl spray nozzle of flow-number 6 and a nominal spray angle of 80 deg with pressure differential and ambient pressure of a nominally quiescent atmosphere.

Although swirl sprays are in theory axisymmetrical, in practice all nozzles are tested for symmetry by a patternation test, a reduction of 20 per cent in the maximum amount deposited radially being acceptable for most combustion applications.

The spatial dispersion of the drops depends very much on the flow pattern of secondary air round the atomizer and observations which do not take this effect into account are of little practical use.
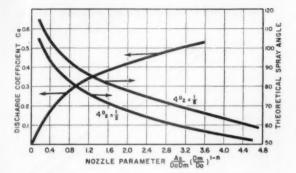
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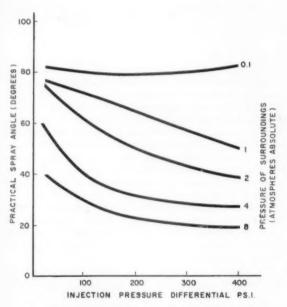Fig. 14—Theoretical spray angle and discharge coefficient for swirl spray nozzle

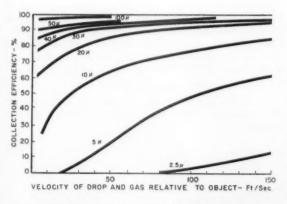Fig. 15—Variation of practical spray angle of a swirl spray

Fig. 16—Collection efficiencies of 1 in. wide flat object

DROP SIZE

The mean size and variation in size are one of the most important performance criteria of sprays. The problem of assessment and analysis is a very formidable one as the size range involves a fifty-fold variation, and the quantities to be dealt with are often hundreds of millions of drops per second (1 cm³ of liquid atomized by a pressure atomizer of flow-number 10 gives 10,000,000 drops).

Practically all estimations are carried out on non-evaporating sprays, which is not only convenient but also justified in the combustion field since, for an assessment of the mass transfer from the spray, the sizes of the initial drops as formed from the spray sheet at the point of disintegration are required. Although the life of the drops depends on the combustion conditions in the chamber, the resident drop size is related to the initial drop size as discussed under "MASS TRANSFER", July issue, p. 22.

Methods of Drop Size Determination. The numerous methods have been reviewed adequately (25); they are based on direct or indirect techniques in which the spray is either sampled or the total spray is analyzed.

Among the indirect techniques used on a large scale, the freezing technique is by far the most frequently applied. The so-called "substitute method" (27) has been developed whereby the fuel properties (particularly viscosity) are simulated by wax which is sprayed as a liquid and solidifies in transit through a cool atmosphere. The end product is a particulate solid mass which can be analyzed for size by sieving. The results from this method can be used for comparison only, for although the properties of the original liquid can be simulated, the solidification process to which the wax spray is exposed can hardly be likened to the evaporation process which takes place on the resident spray in a combustion chamber. Results which can be interpreted more satisfactorily are obtained by freezing out the spray of actual fuel after the zone of disintegration from the sheet. This method has recently been used on a large scale for testing the performance of atomizers for heavy fuel oil, the spray being frozen in a mixture of ethyl alcohol and cardice.

Sampling methods which result in information on drop size in a localized area of the spray are more versatile but suffer from the uncertainty of catching satisfactorily representative samples. In most methods the drops impinge on an object under aerodynamic conditions which might allow the object's impingement characteristics to be calculated. Smaller drops tend to follow flow-lines round the object instead of impacting on it, while larger drops are not deflected because of their greater inertia. The impingement characteristics of objects is expressed in terms of a collection efficiency which is defined as the ratio of the number of drops impacting on the object to the number approaching it in the stream tube swept by the object. Calculations have so far been carried out for drops moving with the same velocity as the surrounding gas, for drag forces which obey Stokes' law, and for flow patterns round the object which are either potential or viscous (29). For a flat object, 1 in. wide, in potential flow, the collection efficiencies shown in Fig. 16 have been constructed. It is seen that for a velocity of approach of 10 fps, 82 per cent of 30μ drops or 98 per cent of 100μ drops reach the object. In general, efficiencies increase with increasing
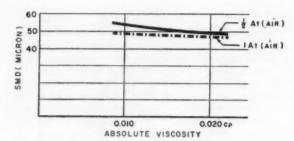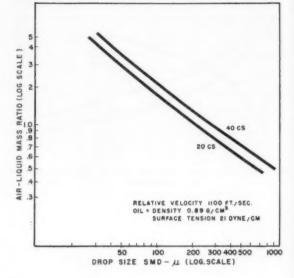
Fig. 17—Effect of viscosity of atmosphere on drop size

Fig. 18—Drop size of sprays from a venturi-type twin fluid atomizer

size and velocity of drops and with decreasing width of object. The aerodynamic conditions for sampling hot or burning sprays are such that efficiencies are likely to be improved relative to the corresponding cold sprays. By means of such efficiency charts, the analysis of a collected sample of drops can be corrected to that of the spray at the location of sampling.

Drop Size Analysis. Since the sizes of drops are within a distinct range say (10–500 μ), they can be grouped according to a frequency distribution. It is the "mean" of this distribution and the "size dispersion" round the mean which are used to express the drop size of a spray. Many attempts have been made to find mathematical functions for the size distributions in order to employ quick graphical methods to assess the mean and the dispersion from straight lines (28). None of these are adequate, and the indiscriminate application of one preferred distribution is open to the abuse of quick interpretation of the position and slope of the line resulting in unjustifiably large inaccuracies. The computing method of analysis consists in forming the required algebraic relation for the particles of one size group and summing for all groups. For instance, the surface mean diameter (SMD) and mean evaporation diameter (MED) is given by the following expressions

$$\text{SMD} = \frac{1}{\Sigma \ (\Delta \bar{v}/d)} \tag{13}$$

$$\text{MED} = \sqrt{\frac{1}{\Sigma \ (\Delta \bar{v}/d^2)}} \tag{14}$$

where $\Delta \bar{v}$ = fractional volume or mass within size group $d \pm (\Delta d/2)$. The surface mean diameter or mean evaporation diameter is defined as the diameter of a monosize spray having the same specific surface or specific evaporation, respectively, as that of the actual space.

While it is common practice to employ SMD as a measure of the mean size, it has been shown earlier that a better criterion for the mean size of sprays undergoing mass transfer under combustion conditions is the MED.

The diameter below or above which is 50 per cent of the mass or volume of the spray on a cumulative distribution, i.e., the mass median diameter (MMD), can be read directly from a distribution diagram. Two other diameters, viz., the 95 and 5 per cent diameters, are used to give an indication of the size dispersion. From the mean diameter of sprays, comparative combustion efficiencies of chambers can be inferred, while the 95 per cent diameter gives an indication of the extent of chamber required to burn the large drops, and the 5 per cent diameter a measure of the ignitability of the fuel sprays. Little quantitative general information is available on these criteria beyond practical figures which have been gained from experience for each particular furnace application.

Performance of Sprays. The initial drop size at the point of disintegration from the sheet depends on the type of atomizer and design of nozzle, the physical properties of the liquid and that of the ambient atmosphere. Many empirical equations for drop size have been published, some of which are summarized in reference (24).

It must be noted that in combustion, most atomizers are twin-fluid devices. Only pressure atomizers are sometimes used either completely without a second fluid for atomization or such that this fluid has little effect on atomization but only serves to direct the spray. Rotary atomizers are certainly always used as twin fluid devices but no specific information on the drop size under such conditions has so far been published.

Pressure Atomizers

It can be shown by dimensional analysis and from first principles that the following general relation applies.

$$d \propto \left(\frac{FN \times \gamma}{\theta \times \Delta p} \right)^{1/3} \rho_a^{\ 1/6} \tag{15}$$

where

d = mean drop diameter of spray
θ = spray angle
γ = surface tension of liquid

For a particular atomizer, and different liquids of the same surface tension:

$$d \propto \left(\frac{C_Q}{\Delta p}\right)^{1/3} \qquad (16)$$

The effect of viscosity (γ_e) is reflected in the discharge coefficient. For fan spray nozzles, C_Q is fairly constant and d may be taken to be independent of viscosity; in fact it can be shown that the effect cannot be greater than $\gamma e^{0.1}$ (ref. (28), Hasson). For swirl spray nozzles, the effect of viscosity on discharge coefficient is appreciable. Under constant pressure operations, a decrease in viscosity will bring about an increase in air core diameter, i.e., a decrease in sheet thickness and a decrease in drop size. Under constant output operations, a decrease in viscosity will increase the tangential velocity which will reduce the pressure differential and bring about a decrease in drop size. If the air core diameter is also affected, the decrease in drop size will be the more pronounced. Because of these interacting factors, the effect of viscosity on drop size has only been assessed empirically. An effect on the order of $\gamma_e^{0.2}$ has been used for swirl spray nozzles spraying light oils.

Although the effect of pressure differential is $\Delta p^{-1/2}$, it has been observed that an increase in pressure differential beyond 200 psi appears to have little effect on the drop size.
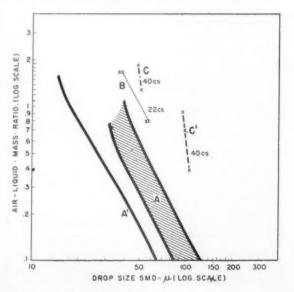
For a particular liquid and surroundings, the drop size is proportional to $(FN/\theta)^{1/2}$ and will thus be affected by diameter of orifice.

For fan spray nozzles,

$$d \propto D_e{}^{2/3} \qquad (17)$$

(where D_e is the diameter of the equivalent circular orifice area) while for swirl spray nozzles, empirical equations have to be resorted to because of the variation of θ with the diameter of the orifice. It has been found that the effect is between $D_0{}^{1.0}$ and $D_0{}^{1.6}$, where D_0 is the diameter of the orifice.

The effect of density of the ambient atmosphere is of the order of $p_g{}^{-1/4}$. There is little knowledge on the effect of viscosity of atmosphere on drop size. For high pressure injection nozzles spraying into cold gases, the effect of viscosity is shown in Fig. 17 (30). The maximum absolute or kinematic viscosity employed in these experiments is about $1/2$ or $1/4$, respectively, of what can be expected under combustion conditions but, from the trend of the results, it is inferred that the effect of viscosity is very small.

The effect of flame surrounding the sheet has been observed to result in a sheet disintegration similar to that occurring in vacuum (31). An increased drop size is expected since the surounding atmosphere will have a greatly decreased density which will bring about an increase in drop size, while the increased kinematic viscosity should have no effect on drop size. These observations apply to the initial drop size at the point of disintegration from the sheet; shatter of drops in a hot atmosphere has always been referred to, p. 27, July issue.

Twin Fluid Atomizers

Extensive investigations on the drop size of sprays from twin fluid atomizers have only been carried out with an internal mixer of the venturi type (see Fig. 10c). There are indications, however, that the trends observed
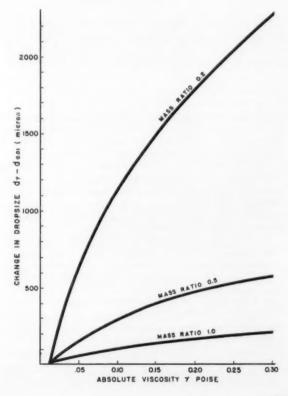
Fig. 19—Effect of liquid viscosity on drop size (venturi-type atomizer)

Fig. 20—Comparative drop size of sprays from twin fluid atomizers. Letters refer to atomizers (Table 4). Figures refer to viscosity

TABLE 4—OPERATING CONDITIONS FOR TWIN FLUID ATOMIZERS TO WHICH RESULTS OF FIG. 20 APPLY

Symbol →	A	A′	B	C	C′
Atomizer → Type →	Internal Mixing		BISRA Nozzle (Fig. 10b)	External Mixing	
	NGTE Nozzle (Fig. 10a)			Vortex Cup	
				Fig. 10d	Fig. 10e
Diameter of orifice, in.	0.25	0.125	0.375	1.0	2.0
Liquid flow-rate, lb/hr	50–270	. . .	200–400	170–230	170–230
Liquid pressure, psi	20–100	. . .	10–50	18–34	1–1.6
Air pressure, psi	10–50	. . .	. . .	10	10
Reference	(32)		(33)	Shell (private communication)	

for sprays from the atomizer are in general applicable to sprays from rapidly moving sheets which disintegrate as a result of gas impingement. The following equation for SMD was found for this design using air, and liquids of viscosity range 0.01 to 0.3 poise

$$SMD = \frac{585}{V}\sqrt{\frac{\gamma}{\rho_e}} + \frac{725}{\rho_e{}^{1.73}}\left(\frac{\eta}{\sqrt{\gamma}}\right)^{0.45}\frac{1}{MR^{1.5}} \quad (18)$$

where

SMD = surface mean diameter (micron)
V = velocity of gas relative to liquid at nozzle exit (m/sec)
γ = surface tension (dynes/cm)
ρ_e = liquid density (g/cm³)
η = absolute viscosity (poise)
MR = mass ratio of air to liquid, with air at 100 F

For a fuel oil of surface tension 21 and density 0.89, the above equation reduces to the following

$$SMD = \frac{2840}{V} + 446\frac{\eta^{0.45}}{MR^{1.5}} \quad (19)$$

This equation is plotted in Fig. 18, and it is seen that the mass ratio is the predominant variable. The plotted results apply to sonic velocity of air at 100 F and a pressure differential of the liquid of 10 psi. The effect of viscosity of the liquid is demonstrated in Fig. 19 where the change in drop size is shown for sprays from liquids at various viscosities relative to that having a viscosity of 1 cp, all other operating conditions viz mass ratio and velocity remaining unaltered. It is seen that the effect of viscosity is appreciable at low mass ratios.

Some results on drop size from twin fluid atomizers used in combustion appliances are shown as a graph in Fig. 20. They are comparative since they were obtained by the same method of drop size determination, viz., the wax substitute method. The operating conditions. design the size of atomizer were all different; the relevant information is given in Table V.

It is seen from Fig. 20 that all five atomizers give a satisfactory performance since their SMD drop size is below 100 μ. Systematic tests have been published only for the NGTE, nozzle A and A′ and adequate data for other atomizers are not available for a general comparison.

Acknowledgments

Editor's Note: Complete List of References appeared in the July issue Part I, p. 29.

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Incinerator Furnace Temperature . . .

How to Calculate and Control it*

By H. G. MEISSNER†

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The recommended procedure and factors of primary importance in the design and operation of a refuse incinerator are offered so as to enable the calculation and maintenance of the optimum furnace temperature. The procedure has been simplified to accomplish the desired results with a minimum of labor and within commercial limits of accuracy.

THE three "T's" of combustion, namely "Time," "Temperature" and "Turbulence" are so closely related that they must be considered together in determining the performance of the incinerator. Without time enough for the combustible matter to burn, the desired temperature will not be obtained, and without adequate turbulence neither the time or temperature requirements will be achieved.

For any given furnace and fuel there is a fairly definite and rather easily determined optimum furnace temperature which can be calculated, when certain controlling factors are known or assumed. Such calculations have been used for many years in the design of boiler furnaces but are not so well known or understood in the case of incinerator design.

These factors include the fuel characteristics such as ash fusion temperature and moisture content for solid fuels; also unit heat value, as well as the furnace size and design, water cooling or refractory wall construction, including boilers or other heat absorbing or "black" surfaces. The S/V ratio or heat absorbing surface vs. furnace volume, is sometimes used to express the latter factor.

The principles involved are the same for any fuel, but we will go through a typical calculation for an incinerator designed to burn municipal or industrial refuse. The moisture content in a solid fuel such as refuse, cannot be readily measured or controlled. However, the total air flow is quite easily regulated, and is the major factor

in carrying away the heat generated, and controlling the furnace temperature, as the attached heat balance calculations and charts illustrate.

The maximum furnace temperature in any combustion is of course obtained when the theoretical air only is supplied. In the case of most fuels including refuse this temperature will be well above 3000 F, which greatly exceeds the allowable value for practically all solid fuels because of ash fusion and clinkering properties.

Excess Air

The excess air introduced into the furnace has therefore been used for years to directly control the furnace exit gas temperature. For most incinerators this has required from 100 per cent to 200 per cent excess, for the all-refractory furnaces normally used, with substantially less air when water cooling is provided. Possible advantages of water cooling will be considered in subsequent sections of this paper.

The radiation loss seldom averages more than 1- to 3-per cent of total heat input and depends largely on the furnace wall construction and temperature differential between furnace and ambient air. Heat absorbed in the brickwork of the furnace before stable conditions are obtained is a function of the weight of brick involved, temperature rise, and specific heat of the brick and seldom exceeds 5 per cent even when starting with a cold furnace.

The hottest part of any furnace is in or near the top of the flame, where combustion to CO_2 has been completed. The flame itself is therefore the hottest arch possible and should be permitted to burn out directly above the refuse on the grate to assure rapid drying and ignition.

When a pound of carbon burns only to CO, it liberates 4450 Btu or about $1/3$ of its total heat the latter being liberated only when the carbon burns to CO_2.

Furnace size has a rather secondary effect on the temperature obtained, as it is possible to obtain practically the same flame temperature over a 2 to 1 or more range in heat release rates. That is, for a given burning capacity the furnace size can vary quite widely, without material effect on the temperature, provided the excess air is carefully controlled.

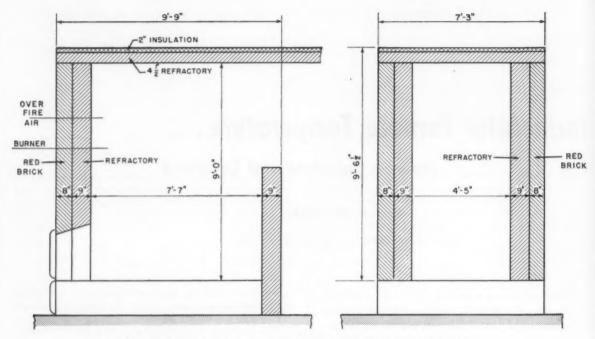All fuel, whether solid, liquid or gaseous, actually

Fig. 1—Sectional views of the typical direct fed type incinerator with again typical dimensions

burns largely as a gas, the heat from the furnace serving to dry out the moisture and volatilize the combustible matter from the fuel bed.

The following discussions and tabulations illustrate the above points for a typical size of incinerator of the direct fed type, Fig. 1.

It should be noted that complete combustion of cellulose fuels such as refuse can be accomplished with 50 per cent or less excess as is the case in the installation where waterwalls have been installed, either with or without accompanying boilers. There is therefore plenty of leeway in the use of cooling air, with the general understanding that the least possible excess air should be supplied compatible with furnace maintenance, to avoid undue cooling, and excessive gas velocities, which directly affect fly ash or particulate emission.

There are three ways only for the heat generated by the burning refuse to escape from the furnace as outlined below.

1. It can be carried away in the combustion products, which include the air admitted under and over the fuel bed as well as the weight of the combustible part of the refuse and its moisture content.

2. Some of the heat will be absorbed in heating up the brickwork during the initial start up, and lost by radiation from the outside walls of the furnace, when equilibrium has been established.

3. The remainder will be lost by air leakage through cracks in the setting and around doors, and unburned hot residue which drops into the ash pit.

In a heat balance as in the army, all heat units must be present or accounted for, hence at the end of such tabulation, there is usually an item covering the difference between the calculated losses and the total heat input from the burning refuse. When there is no such

excess of input over losses, then we can expect the furnace temperature to drop below the desired minimum.

In the determination of any form of heat transfer we are constantly using the specific heats of the various mediums involved, so a brief discussion of this factor is believed desirable.

Specific heat or the Btu required to raise the temperature of 1 lb of a substance 1 deg F is a valuable tool, too often overlooked or misunderstood by incinerator designers. Yet its vital function is well illustrated in the heat balance, Table I which shows what happens to the heat generated in the furnace by the burning refuse or auxiliary fuel.

This heat is never lost or dissipated as is sometimes assumed by inexperienced designers. In order to cool a gas, for example, the excess heat must be transferred to some other medium, such as air, refractory material or water walls.

Every medium, whether solid, liquid or gas, has its specific heat value, which is dependent on the temperature and pressure involved. Engineering handbooks give the specific heats of the products of combustion commonly experienced, as well as of the furnace materials used. These values are of course compared to that of water which is taken as 1.0 or unity, and are taken at atmospheric pressure.

For our purpose the specific heat may be considered as the ability of the various media involved to absorb and carry away the heat generated by the incineration process, which heat may be transferred later to some other medium, for instance, by means of a heat exchanger or boiler.

Water, with a specific heat of 1.0 will pick up and carry out 4 times as many Btu per lb, as will air, having a specific heat of 0.25. This is one reason why water is

TABLE I—HEAT BALANCE—REFUSE BURNING

Refuse assumed to have 20% moisture and 6000 Btu per lb as fired
Refuse burned per hr—1000 lb
Burned with 200% excess air

Gas temperature in furnace— Ambient temperature—70 F
1600 F

Heat loss:

In combustion products	5,430,000 Btu	90.5% of input
By radiation through walls	156,000 Btu	2.6% of input
By Air leakage through walls	180,000 Btu	3.0% of input
From unburned combustible and unaccounted	234,000 Btu	3.9% of input
Total heat losses	6,000,000	100.0%
Total heat input 1000 × 6000	6,000,000	

such a popular heating and cooling medium, and why
it is sometimes used instead of air for reducing the gas
temperature from incinerator furnaces.

When we increase this excess air to say 250 per cent,
the heat loss in the combustion products alone increases
to 6,350,000 Btu per hr, which combined with the other
losses, adds up to 6,920,000 Btu, or considerably more
than the total heat input, so that the furnace tempera-
ture drops, as shown in Fig. 2.

The radiation loss of 2.6 per cent can be reduced by
use of an insulated setting, to perhaps $1\frac{1}{2}$ per cent, or
increased to 4- or 5-per cent if the setting is reduced in
thickness. In the latter case the outside wall tempera-
ture would be so high that the operators would be en-
dangered. The close relationship between furnace con-
struction and outside temperature is clearly illustrated in
Table II.

It is evident from the above heat balance that by far
the greatest and most easily controlled loss is that due
to the air flow into and through the furnace. We cannot
change the radiation loss without rebuilding the furnace,
and the moisture loss is determined by the refuse itself,
which generally must be taken as it comes. Hence the
emphasis for good incineration must be put on the fuel-
air ratio, or the amount of combustion air admitted to
the furnace in a given time.

In the above example the heat loss by transfer to the
moisture in the refuse, reached to 362,000 Btu per hr,
or 6.05 per cent of the total heat input. Control of this
moisture and resultant heat loss is desirable but usually
difficult or impossible, hence as noted above the most
practical way to regulate furnace temperature is by
means of the entering combustion air.

Contrary to popular belief, the major function of the
walls and roof of the furnace, is not to reflect the heat

TABLE II—EFFECT OF FURNACE WALL CONSTRUCTION ON OUTSIDE OR COLD FACE AND TEMPERATURE RADIATION LOSS WITH 1600 F FURNACE TEMPERATURE

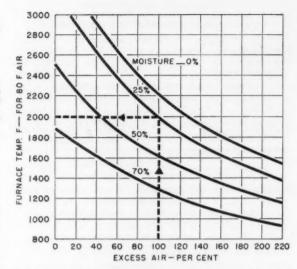| Wall Construction	Cold Face Temperature, F	Radiation Loss Btu/sq ft/hr	Loss in Per Cent
$4\frac{1}{2}$-in. plastic refractory	440	1356	6.5
9-in. plastic refractory	440	1356	6.5
$4\frac{1}{2}$-in. plastic refractory + 1-in. insulation	262	495	2.4
8-in. plastic-refractory + 4-in. insulation	153	150	0.72
9-in. firebrick, + 8-in. red brick	274	545	2.6
Water walls, $2\frac{1}{8}$-in. insulation	140	138	0.66

Fig. 2—Anticipated incinerator temperatures and the air required to control them can be determined from this graph

back into the furnace, but to direct the gas flow and
prevent the admission of uncontrolled cold air. Even
in a refractory furnace, the wall temperature is always
somewhat cooler than the flame temperature, except
perhaps for a few moments after the fire is cleaned or
the heat input is otherwise suddenly reduced. Heat will
flow only from high to low temperature, and unless the
walls are at a higher temperature they cannot reflect
heat back to the flame.

This was proved definitely many years ago, with the
advent of water wall furnaces, many of which now have
no refractory at all next to the fire, yet such furnaces
can be and are operated at higher temperatures because
there is much less trouble with slagging or refractory
maintenance.

Similarly the heat-up time required is affected much
less by the furnace wall construction than by the control
of the fuel-air ratio during this period. The belief
that the heat absorption by the cold refractory furnace
walls materially delays the obtaining of full furnace tem-
perature is also refuted by experience with water wall
furnaces, in which the heat absorption per sq ft per hr is
at a much faster rate—possibly 30 times—and which
never get above the water temperature of several hundred
degrees, F, yet in practice such furnaces reach full op-
erating temperature within a comparatively short time.
This time interval is usually determined by allowable
expansion stresses rather than firing rate as any ex-
perienced operator knows.

It is true that a considerable amount of heat is ab-
sorbed by the refractory walls in a cold start, and that
it maybe several hours before full wall temperature is
reached. Calculations show that about 8,600,000 Btu
are absorbed by the refractory in the example furnace,
but it may take ten hours or more for this heat loss from
the furnace to occur, during which time the excess air
flow, and resultant heat loss in the combustion products,
may be correspondingly reduced.

Radiation losses do not start until the outside wall tem-
perature has risen appreciably above the ambient air,
so that such losses are negligible during the warm up
period.

TABLE III—CALCULATION OF FLAME OR FURNACE TEMPERATURE FOR REFUSE INCINERATOR

(See Fig. 1 for Furnace Construction)

Refuse at 6000 Btu per lb as fired
Moisture....................................20 Per cent
Non-combustible............................. 5 Per cent

Ultimate Analysis of Typical Refuse, Per Cent

Carbon	Hydrogen	Oxygen	Nitrogen	Sulfur
50.0	6.0	40.0	3.0	1.0

Excess air at flame port..............200 Per cent
Combustion products................2040 lb per 1,000,000 Btu
heat input (gross)
Ambient air temperature.............80 F

$$\text{Furnace temperature} - T = \frac{\text{Btu input (net)}}{Wg \times Spm} + 80 \text{ F}$$

Btu input (net) = Btu input (gross) × correction factor K per Fig. 3

Wg = combustion products — lb per million Btu
Spm = mean specific heat above 80 F

(T − t) = gas temperature rise above ambient
$$Wg \times (T - t) \times Spm = \text{Btu input (gross)} \times K$$
$$(T - t) = \frac{\text{Btu input (gross)} \times K}{Wg \times Spm}$$

For above conditions

$$(T - t)\frac{1,000,000 \times 0.805}{2040 \times 0.28} = 1410 \text{ F}$$
$$T = 1410 + 80 = 1490 \text{ F}$$

TABLE IV—HEAT BALANCE—REFUSE BURNING

(Effect of Oversize furnace—Double Normal Size)

Refuse assumed to have...............20 Per cent moisture
Refuse assumed to have...............6000 Btu per lb
Refuse burned per hr.................500 lb
Refuse burned with..................200% excess air

Gas temperature in furnace............1600 F
Ambient...............................70 F
Total heat input......................500 × 6000 = 3,000,000 Btu/hr

Heat lost:
In combustion products.............2,715,000 Btu/hr
By radiation through walls......... 156,000 Btu/hr
By air leakage through walls....... 180,000 Btu/hr
From unburned combustible and
unaccounted.................... 117,000 Btu/hr
_____
3,168,000

Total heat input, per above...........500 lb × 6000 Btu = 3,000,000 Btu/hr

Furnace temperature will decline as input is less than output with above excess air.

Ditto with excess air reduced to.......150 Per cent
Heat lost:
In combustion products.............2,300,000 Btu/hr
By Radiation through walls......... 156,000
By air leakage through walls....... 180,000
From unburned combustible and
unaccounted.................... 117,000 Btu/hr
_____
Total heat losses.....................2,753,000 Btu
Total heat input per above............3,000,000 Btu

Furnace temperature will hold steady or rise as input is greater than output.

The desirability of closing all doors and dampers to avoid undue cooling of the refractory walls, during idle periods is apparent.

Furnace Considerations

Furnace size and design do of course affect incinerator performance in several ways. When the furnace is too small, combustion cannot be completed until some of the volatile combustible matter has passed over the bridge wall, because of lack of the "Time" factor noted above. The heat-up time for such a furnace can therefore actually be delayed, as the heat generated in the secondary chamber does no good in heating the primary furnace. In the case of a high moisture refuse, this heat loss can seriously reduce the incinerator capacity, as a considerable amount of heat is required to dry out and evaporate this water content.

A check of the Steam Tables will show that it takes 1810 Btu to heat a pound of water from 70 F to 1600 F, or 30 per cent of the 6000 Btu available in our example refuse. Both theory and practice show that at something over 70 per cent moisture in the refuse, the latter ceases to be "auto combustible," and that relatively expensive auxiliary heat is required to maintain the furnace temperature. This is illustrated in Fig. 2 where the theoretical maximum temperature with no excess air or other heat losses is only 1900 F, and at 50 per cent excess air the temperature has already dropped below our 1600 F
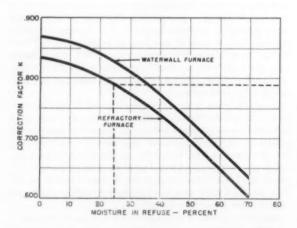

Fig. 3—Correction factors for net vs. gross heat input for the different furnace wall constructions are plotted here and used in conjunction with Table III
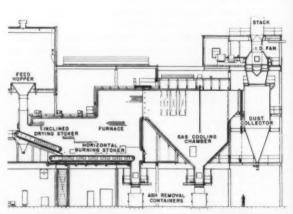

Fig. 4—A large municipal incinerator using a traveling grate is produced here for contrast with the typical unit of Fig. 1

level. Table III and Fig. 3 show calculations involved.

The effect of an oversize furnace is easily shown by assuming that the amount of refuse burned per hr is reduced by half together with the combustion air introduced into the furnace. The radiation loss will remain the same, assuming that the furnace temperature is maintained, and air leakage through the brickwork and around doors will also be unchanged. When these constant losses, added to the loss in the combustion products, are greater than the heat input, then the furnace temperature will drop, and incineration efficiency will drop unless the excess air is reduced as the following example illustrates. See Table IV.

While it is generally easier to obtain good performance with an oversize furnace as compared with one which is too small, there are several factors which limit the desirable size for a given capacity. Perhaps foremost is the cost, which would normally affect materially the incinerator selection. It is also more difficult to avoid air leakage through the walls and doors of an oversize furnace, and the radiation loss will of course be higher because of the greater wall area.

In the very large boiler furnaces now common, the entire furnace is made practically air tight by means of welded steel tubes and casings, with heavy layers of insulation to reduce radiation loss. Many of the new boilers are designed and operated with a positive pressure in the furnace, to further reduce the infiltration of cold air, as well as to eliminate the need for an induced draft fan.

Such construction details would be much too costly, or otherwise impractical for an incinerator, so the design should be proportioned to obtain the optimum performance for the required capacity at minimum cost.

Automatic Control Equipment

Automatic combustion control equipment is very helpful in maintaining the desired furnace temperature, as the temperature sensing device can be used to regulate the total air flow to the furnace, in accordance with requirements.

For example, if the temperature starts to fall, the total air flow will be reduced by means of the forced draft fan damper, until the temperature is stabilized at the desired point. The reverse action will be taken as soon as the temperature rises unduly, and this regulation can be accomplished much more quickly than the average operator would be likely to react.

Such controls should also incorporate a barometric or other damper to maintain the desired low furnace draft at all times to minimize the air leakage into the furnace.

Care in the use of the charging door, and in raking the fire, as well as in cleaning out the residue, is also beneficial, as the rush of cold air into the furnace is very detrimental, lowering the gas temperature and chilling the brickwork. The development of automatic feeding devices to facilitate charging at a uniform rate, preferably without having to open the charging door, is recommended.

The large municipal incinerators using traveling grate or similar stokers, wherein the refuse hoppers are kept full by the operator, and the stoker movement controls the feed rate, show what can be done in this respect. See Fig. 4.

The skill of the operator is still a major factor as regards satisfactory incineration, and a proper training and supervising program is essential.

Installation of Packaged Steam Mains

The cost of steam main installations according to Pierre C. Boquel, Consolidated Edison Co. of N. Y. is a major item in the expansion of the district heating industry. Late in 1959 the author's company formed a Steam Distribution Committee with engineering and construction personnel to study the problem of cost reduction. Materials used in the 1955 design were reviewed and significant savings were realized by reducing the thickness of the steel pipe conduit, by substituting weld end expansion joints for flanged end joints, precast concrete manholes for field poured manholes, and weld end valves for flanged valves. Analysis showed installation represented about 65 per cent of the total cost and that field fabrication was a high item, therefore the major emphasis of this paper describes the Committee's decision to establish packaged items of the major components and reduce field fabrication to less than 10 per cent of the total work of installation.

A contractor can now bid a job knowing that time consuming field fabrication will be limited in scope; that precast concrete manholes can be installed without waiting for above freezing temperature; and that work yards and storeyards are not required because the Contractor can order the packaged units to be delivered on the job to match his trenching progress.

The designs of the packaged units were established by the following conditions:

1—Prefabricated pipe units are as long as trench obstructions will permit.

2—Expansion joint units are based on the use of weld and expansion joints.

3—Manholes have minimum working space to limit weights.

4—Anchors are designed to function without field poured concrete.

Pipe Lengths

The pipe unit consists of the pressure pipe, thermal insulation, pipe support rollers, and the conduit pipe. Experience has established 40 ft as the longest single length that can be efficiently lowered in the trench. After annual requirements are known, orders are placed for pipe, insulation and roller supports and this material is sent directly to a prefabricator for assembly and storage until required. The prefabricator, generally a pipe coating company, applies a hot line enamel on the casing pipe for corrosion protection, and provides the labor to assemble the parts. The insulation,

ordered in full round cylinders 3 ft long, can be slipped on without any cutting and without the banding required for half round sections; thirteen 3 ft lengths are required for each 40 ft unit of pipe. It is hoped that ultimately this unit can be ordered complete from one vendor; however, the present method of ordering costs less

Expansion Joints

The unit consists of a packless type expansion joint with welding ends, thermal insulation, and a steel casing, with integral pipe guides, that provides a watertight enclosure when welded into the main conduit. The expansion joint manufacturers ship the unit, completely ready to install. This is a rugged unit because the corrugated elements of the joint is protected from damage during installation. It is pre-sized to the exact installed length. Handling and line up can be done with only the care needed to install pipe units. Availability of joint units is provided by placing one order for the annual needs and requiring the manufacturer to maintain a minimum stock of all sizes. This design could be used for flanged end joints, by using a larger steel casing, however the possibility of gasket leaks must be considered.

Concrete Manholes

The unit consists of the precast concrete floor and walls; pipe, valves, fittings, thermal insulation and anchors completely installed; and a precast concrete roof.

The manholes are cast with short lengths of conduit pipe in the end walls to permit the manhole piping to extend beyond the walls, the roof slab has all the openings required for entrance heads and valve operating rods.

When the manhole arrives on the job, it is lifted off the truck, placed in the trench, lined up and welded to the adjacent pipe units, using the same material and method required for field welding all other units together. The roof is set in place on a 1 in. bed of mortar; the manhole castings are set to grade; and the trench is ready for backfilling.

The largest packaged manhole installed to date if 8 ft long, 5 ft wide, 5½ ft high; and weighs 23,800 lb including the equipment; the roof weighs 5700 lb. Manholes are poured with the forms inverted to prevent porosity in the floor and at the floor and walls, insuring a water tight structure. Manholes for steam traps are uniform in size 4 ft long, 3 ft wide and 3½ ft high and are precast with floor, walls and roof in one piece. The traps and related piping are installed before delivery to the job.

More than 500 utility boilers have been cleaned using thiourea in a formulated hydrochloric acid solvent, a process covered in U. S. Patent 2,959,555. Until the time that patent protection was established (Nov. 1960) the manufacturer understandably withheld publication. Here is a brief account of the chemistry involved and certain of the early case histories.

By J. P. ENGLE

Dow Industrial Service

The Merits of a Single Stage Solvent for Simultaneous Removal of Copper and Iron Oxides

SINCE copper does not normally displace hydrogen, even from acid solutions, it is virtually unattacked in most non-oxidizing solutions. Most acid solvents used in chemical cleaning are of the non-oxidizing type, therefore, it would seem wise to consider how copper may be brought into solution in acids such as hydrochloric acid solutions. Perhaps some single stage solvent could be developed. This possibility seemed so attractive we decided to set about finding such a solvent.

The iron oxides, normally present in boilers prior to cleaning, may have the chemical composition of FeO, Fe_2O_3 or Fe_3O_4. Any of these compounds may also exist in combinations. Upon dissolution of these oxides in acids, ferrous or ferric ions or both may be formed as follows under boiler cleaning conditions.

$$FeO + 2HCl \text{---)} FeCl_2 + H_2O \text{ (ferrous ion, } Fe^{++}) \quad (1)$$

$$Fe_2O_3 + 6HCl \text{---)}$$
$$2FeCl_3 + 3H_2O \text{ (ferris ions, } Fe^{+++}) \quad (2)$$

$$Fe_3O_4 + 8HCl \text{---)} 2FeCl_3 + FeCl_2 + 4H_2O$$
$$\text{(both ferrous and ferric ions)} \quad (3)$$

Drain samples taken from boilers fail to show the presence of ferric iron in amounts sufficient to satisfy equations (2) and (3) above. At the same time copper may be present in solution. The absence of ferric iron and the presence of copper in solution is explained by equation (4) and (5) below.

$$2Fe^{+++} + 2Cu° \text{---)} 2Fe^{++} + 2Cu^+ \quad (4)$$

$$2Fe^{+++} + Fe° \text{---)} 3Fe^{++} \quad (5$$

The $FeCl_3$ of equation (2) or (3) is seen to react with copper in the deposit to form a soluble copper salt. This reaction is quantitative in character and rapidly attains equilibrium favoring reduction of the ferric iron with resultant dissolution of metallic copper.

It has been observed in many boilers that unless the copper is removed in a separate treatment prior to removal of the iron oxides with acid solvents, the interior metal surfaces are plated with copper during the acid stage. This is in accordance with equation (6) below.

$$2Cu^+ + Fe° \text{---)} Fe^{++} + 2Cu° \quad (6)$$

Equations (4) and (5) above indicate that the ferric ion is corrosive and will dissolve stoichiometric amounts of copper or iron. If we assume 2000 lb of magnetite to be present in a boiler, the ferric ion content of this material will dissolve 482 lb of metallic iron from the boiler by reduction of the ferric ion to ferrous. Likewise, equation (6) tells us that if we allow 200 lb of copper to plate in the boiler (which must subsequently be removed), we have dissolved an additional 88 lb of boiler steel.

On the other hand, the most important finding of all has thus far not been mentioned. We cannot prevent formation of the ferric ion in accordance with equations (2) and (3) during cleaning operations. We can use equation (4) to advantage to dissolve metallic copper which is ordinarily acid insoluble. In so dissolving the metallic copper, we have decreased the loss in boiler metal by an equivalent stoichiometric number of pounds. If we can prevent copper plating, we have further decreased the loss in boiler metal. And this is precisely what thiourea does in hydrochloric acid solutions. Thiourea will not dissolve copper in any form; it does form a highly stable copper-acid-complex so that copper plating is prevented, Fig. 1.

Summarizing these technical considerations, we find:

1. Great quantities of ferric ion are formed during cleaning of high pressure utility boilers.

2. The ferric ion will dissolve stoichiometric amounts of boiler steel and/or metallic copper which may be in the deposits.
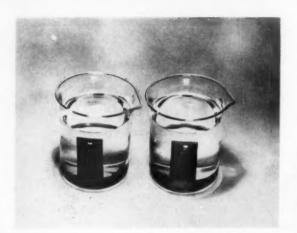
Fig. 1—Prevention of copper plating on steel by addition agents. Left—No additive, note plated copper. Right—copper plating prevented by additive

Fig. 2—Laminated deposit of copper and magnetite. 4 per cent Nital etch, 200X

3. If the copper is complexed as rapidly as it dissolves by the addition of thiourea, we have prevented loss of an equivalent stoichiometric amount of boiler steel.

4. By complexation of the copper, we have prevented metallic copper from plating, thereby again saving the stoichiometric equivalent loss of boiler steel.

The above four items refer to advantages gained solely by a reduction in corrosion by the use of thiourea. The primary advantage is that, both copper and iron oxides are removed in a single stage fill. Silicates and phosphate compounds offer no interference whatsoever and can be effectively removed in the single stage treatment.

Practical Aspects

Theory must be reduced to practice before research "pay its way." Prior to actual tests there was much misgiving and concern as to the outcome. Immediately following the first field application a number of tubes were removed from the high heat input section for examination in addition to a thorough examination of the boiler itself through all accessible openings. The water treatment representative, customer, and our own personnel were present for these examinations. All present agreed that the boiler was clean with no copper plating observed, the uniform metal surfaces showed that neither localized or general thinning type corrosion had been produced. Research had paid off and an expanded sales and development program was in order.

Conservatism on the part of some utility operators caused a request for samples of the then new product for evaluation in their own laboratories. These independent evaluations, without prior consideration of the entire series of chemical reactions involved, created concern

in the minds of a few that the solution was corrosive. This concern was falsely predicated on insufficient data of the type shown in Table I.

No one can deny from the data of Table I that the corrosion rate has been increased 175 per cent simply the addition of the thiourea. The inhibitor in this case, Table I, is highly effective without thiourea as shown by the fact that only 0.024 mils metal penetration occurs during a 6 hr treatment. Prior to the development of improved inhibitors through research, many boilers have been treated at higher corrosion rates than we are now considering, therefore the sudden concern of 3 times nothing appears unfounded in the author's opinion.

If the above causes concern to some, let's take a look at the system as conditions actually exist during cleaning operations. We must add copper and iron oxide (magnetite) to our tests in addition to a weighed-coupon-for-weight-loss studies in order to more closely simulate cleaning conditions. Then let's explore the two systems, namely ammoniacal bromate and hydrochloric acid versus hydrochloric acid containing thiourea with respect to corrosion rate. The data obtained in such a study are shown in Table II. The copper and magnetite were added in quantities sufficient to yield 0.1 per cent copper and 1.5 per cent iron in solution. For ease in tabulation, the ammoniacal bromate is designated M63; thus a solvent system written M63:HCl means that the first stage was ammoniacal bromate followed by 5 per cent inhibitied hydrochloric acid.

Examination and study of data in Table II will show the following conclusions are warranted regarding these

TABLE I—CORROSION RATES—WITH AND WITHOUT THIOUREA

Specimen material—AISI 1010 Cold Rolled Carbon Steel
Exposure—5 Per Cent Hydrochloric Acid plus 0.2 Per Cent A74 Inhibitor, 6 hr, 150 F

	Lb/Ft²/Day	Mils penetration per 6 hr
With Thiourea	0.011	0.067
Without Thiourea	0.004	0.024

TABLE II—CORROSION RATES IN VARIOUS SOLVENT SYSTEMS

Specimen Material—AISI 1010 Cold Rolled Steel
Exposure—Solvent Systems as shown, 6 hr 120 F for M63, 150 F for HCl

Test No.	Solvent System	Corrosion Rate-Pounds per Square Foot per Day	
		No Fe and Cu added	Fe and Cu added
1	M63:HCl	0.007	0.061
2	HCL:M63:HCl	0.014	0.070
3	M63:M63:HCl	0.007	0.062
4	HCl + Thiourea	0.011	0.073

complete solvent systems employed for the dissolution of iron oxide and copper.

1. In Test No. 1, wherein iron oxide and copper were added, the corrosion rate of 0.061 confirms the corrosive nature of the ferric ion formed by dissolution of the magnetite after the removal of copper by the M63 solution.

2. In Test No. 2, the rate of 0.070 is slightly higher because copper plating and ferric ion corrosion are brought into play. The first acid stage yields ferric ion which dissolves the metallic copper which subsequently plates on the test specimen with an overall increase in metal loss.

3. The test again reflects the corrosivity due to ferric iron inasmuch as the copper was previously removed.

4. The rate here of 0.073 is of the same magnitude as Test 2 wherein copper plating was permitted to occur. If this magnitude is such as to cause concern, comparison with Test No. 1, a normally accepted procedure, will reveal a 20 per cent increase in corrosion rate. This is insignificant for the same reasons that the erroneous increase of 175 per cent previously referred to is insignificant.

Thus we find that the conservatism and fear is unjustified on the basis of properly conducted laboratory tests. There is a very important item not brought up for discussion thus far and that is the inhibitor prescribed for use with thiourea in hydrochloric acid solutions. Many inhibitiors are ineffective; others are only partially effective; none to the author's knowledge has been found as effective as one developed and used exclusively by the author's company.

Field Studies

Concurrent with the laboratory research and development programs, the experiences in field practice were closely followed. It was soon found that magnetite matrices containing metallic copper of the type shown in Fig. 2 were rapidly disintegrated and dissolved. The reaction rate was found to be such that virtual dissolution of deposits is completed in the first hour. This is attributed to the conjoint action of solubilization and complexation occuring at the same time.

How effective is the system in practice? This question

TABLE III—FERRIC ION EFFECT

Test No.	Solution	Additive	Corrosion Rate, Lb/Ft²/Day
1	5 per cent HCl	None	0.004
2	5 per cent HCl	0.75 per cent FeCl₃	0.042
3	5 per cent HCl	0.75 per cent FeCl₃ plus 1 per cent thiourea	0.020

was asked of our personnel very shortly after adoption of the new material. At this particular first field installation, chemical analysis of the acid-thiourea drain sample showed 3500 lb of magnetite and 160 lb of copper removed in the single stage treatment. A second fill of acid was made to determine whether deposits remained or not. Analysis of this second stage acid drain showed only 98 lb of magnetite to have been removed. No copper plating was observed upon inspection, therefore it is assumed that all the copper was satisfactorily removed in the first stage.

On several instances, test coupons have been suspended in the boiler drum for exposure during chemical cleaning of the unit. In those cases, wherein the coupons contained deposits of magnetite and copper, the deposits were removed with no evidence of corrosion. In other cases wherein no deposits were on the coupon, weight losses and examination confirmed that corrosion was no greater than expected. These findings are in accordance with previously presented laboratory data. In view of the rapid reaction rate, the major portion of the thiourea is early consumed in forming the tightly associated copper complex. It has been subsequently found that thiourea complexes ferric iron also, thus the remaining thiourea, in excess of that required for copper complexation, is probably in an associated ferric iron complex. This is indicated by a decrease in corrosion rate as shown in Table III.

There appears to be some indication that the regular use of thiourea may play a role in the reduction of tube failures. This is based on examination of our company's role in the reduction of tube failure samples received in the laboratory for examination. The deposits associated with such tube failures are localized, hard, dense, deposits of magnetite which may or may not contain copper.
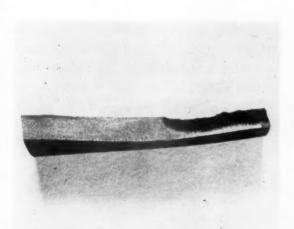
Fig. 3—Longitudinal section of tube after removal of magnetite plug. Dark area is embrittled metal. Hydrochloric acid etch
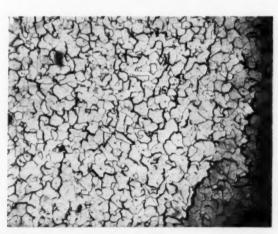
Fig. 4—Appearance of embrittled metal of Fig. 3 at 200X, 4 per cent Nital etch

TABLE III—Fe₃O₄ REMOVED WITHOUT THIOUREA					
Capacity, pph	Volume, gal	Pressure, Psi	Fe₃O₄, lb Removed from new Boiler	Opera- tional Periods Months	Fe₃O₄, lb. Re- moved after Opera- tion
1,300,000	43,000	1800	6400	24	4400
875,000	17,000	1800	1785	14	1310
1,300,000	62,000	1800	7316	29	3300
875,000	?	1800	1650	15	1190

TABLE III—Fe_3O_4 REMOVED WITHOUT THIOUREA

TABLE V—Fe₃O₄ REMOVED WITH THIOUREA PRESENT			
Capacity, pph	Volume, Gal	Fe₃O₄, lb Removed	Cu, lb Removed
600,000	21,400	3420	250
1,000,000	27,225	1980	135
350,000	18,000	3920	94
?	32,000	1850	118
?	30,000	740	112
?	41,500	2580	207
?	33,000	2180	230
?	18,750	980	133

The metal is generally seriously embrittled beneath the deposits as shown by metallographic examination, Figs. 3 and 4. They are certainly not confined to geographic areas as we have examined tubes from installations on the east coast as well as from the west coast. Strangely enough, for unknown reasons, the requests for examination of such tube failures have shown a sharp decrease since about 1958. This constitutes a very intangible and nebulous sort of evidence but it may be of much greater significance than can be factually determined at this time.

Field Aspect of the Problem

Since the initial use of thiourea for complexing copper during boiler cleaning, close records have been maintained on over 100 field jobs. Such solutions have been satisfactorily used for servicing the different types of utility boilers when used with a proper understanding of its limitations. The addition of thiourea to hydrochloric acid does not constitute a solvent for metallic copper; the thiourea simply complexes the dissolved copper. With this knowledge or foresight of the chemical reactions involved, it would not be expected to remove metallic copper from orifices of controlled circulation boilers; in such cases an ammoniacal bromate solution should be used prior to the acid complexing solution. A sufficient amount of iron oxide is not present in these copper-rich deposits to provide dissolution of the metallic copper.

Another limitation of the acid thiourea solution is that it will not clean local areas of abnormally heavy deposits of the type shown in Fig. 5. However, neither does the ammoniacal bromate solution offer assurance that these areas will be cleaned. In fact, there does not appear to be an entirely satisfactory answer to this problem; because of the conjoint and synergistic action of thiourea in hydrochloric acid on copper bearing iron oxide matrices, this is probably the best solution to the problem thus far devised.

The aspect of corrosion has not been confirmed by studies of drain sample reports nor by examination of a great number of boilers after job completion. The amount of iron found in the drain samples cannot be used as a criterion as this appears to be a function of operational characteristics, time between cleanings, and surface area of the unit as examination of Table IV will show. These data are for boilers cleaned without the addition of thiourea.

We do not have data corresponding to these same units in Table IV for cleaning operations with thiourea in hydrolchloric acid. However, a few samples are given in Table V of treatments made with thiourea; they cannot be regarded as representative as the number of pounds of Fe₃O₄ removed during such treatments have varied widely dependent upon the same factors previously

pointed out. Likewise, the amount of copper removed is dependent upon operational conditions.

The variation in Fe₃O₄ content in Table V is between 740, 1850 and 2180 lb for three units of comparatively the same volumes, i.e., 30,000 gal simply indicates that some units are dirtier than others. It cannot be construed to mean that the acid thiourea solution is corrosive, else all three figures would tend to approach a common value instead of the wide spread observed.

This is one other limitation which should be mentioned; however, if the basic mechanisms are thoroughly understood, this problem should not arise. Thiourea forms several complexes with copper, one of which is insoluble. If sufficient thiourea is used, only the highly stable soluble complex is formed; if an insufficient amount is used, the insoluble complex may form. This has not presented a problem of any serious concern in the field. It has been encountered in a few isolated cases where no previous records existed as a basis for proper solvent formulation.

As with any other cleaning service, tool or technique, a certain amount of information is required for proper application of the acid thiourea solvent. A sufficient backlog of field information has now been gained so that the material is now used repeatedly on a routine basis without any difficulty. Indeed, customer acceptance and satisfaction with this technique have been very gratifying to the author's company management. The material underwent a rapid growth and expansion in field development due to unusual customer acceptance. Such rapid growth is not generally associated with products released from research for field development. This in itself constitutes justification for further research into the problems associated with chemical cleaning irrespective of industry type.
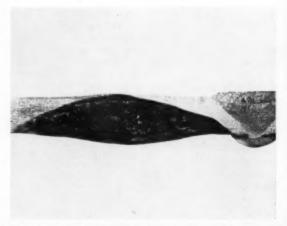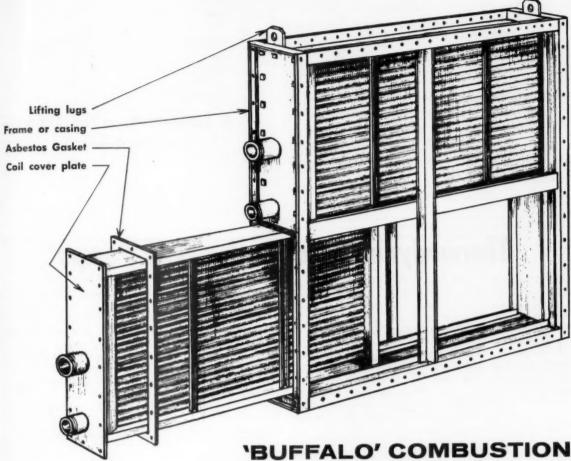
Fig. 5—Cross section of tube wall showing weld back-up ring and penetration of tube wall consumed in formation of magnetite. Hydrochloric acid etch

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The higher pressures of today's power plants make it hazardous to design pumps on the assumption that water is incompressible or specific volumes, densities or heats can be neglected. Here is a brief discussion of some of the influencing considerations

By A. J. STEPANOFF and H. A. STAHL

Ingersoll-Rand Company

Thermodynamics of Boiler Feed Pumps

THE increase of pressure, temperature and power of boiler feed pumps in recent years has presented a number of problems which require familiarity with thermodynamics and water properties at high pressure and temperature for solution. Under the operating conditions of modern boiler feed pumps, water cannot be considered incompressible and variation of water specific volume or density and specific heat cannot be neglected. These effects become more pronounced at low capacities. There is for several reasons a minimum flow for each pump below which pump operation becomes unsafe. Procedure established years ago for calculation of the temperature rise of water in terms of capacity and the minimum flow to avoid cavitation or pump overheating became inaccurate and requires revision.

1. Adiabatic Compression

For determination of changes of water properties as it progresses from the pump suction to the discharge it is convenient to use Ellenwood and Mackay Charts, plate B, p. 24, reference (1),* or, on a larger scale, charts prepared by Tsu and Beecher (2) Fig. 1 shows performance of a 6-in. pump which will be used for the numerical examples to show several states through which boiler feed water passes.

The effect of compression can be found in Fig. 2 by following from point (1) representing inlet pressure, $p_1 = 132.7$ psia, temperature, $T_1 = 331$ F, to the point (2) at the discharge pressure, $p_2 = 3687$ psia, along the constant entropy line, $s = 0.478$. The temperature rise of 5 deg F is observed. The change in specific volume is small.

2. Discharge Temperature

To locate the discharge point (3) on the constant pressure line, $p_2 = 3687$ psia, at capacity 200 gpm the

enthalpy at this point h_3 is found from the enthalpy equation.

$$h_3 - h_1 = w - q \qquad (1)$$

where w is pump work input per pound of water equal to

$$w = \frac{\text{bhp} \times 2545}{W_s} \qquad (2)$$

where W_s is the suction weight of flow per hour equal to

$$W_s = W_d + W_L \qquad (3)$$

where:

W_d = weight of flow at discharge

W_L = weight of drum leakage flow (returned to de-aerator)

q = heat loss from the system by radiation, bearing friction and external seal losses.

All quantities are in Btu's.

The last item q is very small in comparison with bhp and can be omitted from consideration. For 200 gpm, $w = 58.5$ Btu and $h_3 = 360.7$. The temperature is read off from the chart, $T_3 = 381.5$ F, and specific volume, $v_3 = 0.0185$. Contrary to the statements made, (3–5) *the discharge temperature rise has no effect on the suction conditions as to cavitation in the first stage impeller inlet.* For "mechanical reasons" it has been suggested to limit the discharge temperature rise to 20 deg F (6). It will be shown that this figure can be unnecessarily low.

Note that the entire power input (w) contributes to the temperature rise of water at discharge and not only the power lost by internal hydraulic losses, as is frequently assumed (6). This is clear from the Tsu and Beecher diagram in Fig. 2. The hydraulic losses are equal to

$$\text{Losses} = w(1 - e) \qquad (4)$$

where e is the pump efficiency. The corresponding point (2') is shown on the diagram. The difference between the temperatures at point (3) and (2') is 9.3 deg F (at

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* Numbers in parentheses refer to similarly numbered entries in the List of Reference at the close of the article.
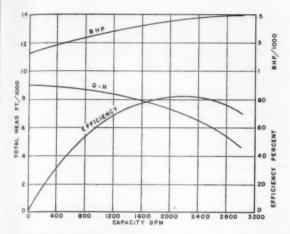
Fig. 1—A typical family of pump performance curves was established above to serve as numerical examples for the text


Fig. 2—The above figure charts the effect of compression for a water at inlet pressure and conditions of ① to discharge at point ② at constant entropy or at point ③ for constant pressure

1200 gpm the difference is 12 deg F). This requires an appreciable increase in the minimum rate of flow to maintain a desired temperature rise, such as the recommended 20 deg F. The difference between the calculated temperature rise based on the heat due to internal losses and actual rise have been observed in the field. In fact, measured water temperature rise from inlet to discharge of boiler feed pumps has been used for determination of the pump power input (w in equation (1)) when other means of measuring input were not available (8).

The increase in specific volume from suction to discharge is 2.5 per cent and is sufficient to show a drop of head in feet near shut-off if the change in water density is neglected. A *pressure-capacity* curve may show a "droop" at very low capacities. This normally occurs at capacities less than the required minimum flow and hence will not complicate governing devices when paralleling two pumps. Note that points (2) and (2') do not represent the actual states of water; the true path of water states from inlet to discharge is not known.

3. Minimum Flow

To limit temperature rise at the pump discharge to a predetermined or assumed value, the flow through the pump should not be reduced below a certain minimum. This is determined from the plot of temperature rise curves (Fig. 3) against capacity. The frequently quoted maximum permissible temperature rise of 20 F was based on the assumption that only heat due to hydraulic losses within the pump contributes to the water temperature rise through the pump. Actually the temperature rise based on the enthalpy increase (enthalpy is a different name for the older term "total heat content") equal to entire pump power input is appreciably higher. On Fig. 3 the minimum flow for 30 deg F rise at pump discharge is 370 gpm based on the enthalpy rise. At the same capacity the temperature rise due to losses only is about 20 deg F. Taking 20 deg F for a true temperature rise would require 600 gpm minimum flow, considerably more than is required to suppress flashing in the balancing chamber as will be shown below.

In general the limit for the temperature rise through
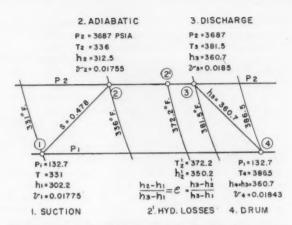
the pump is rather arbitrary. For pumps without balancing devices, such as having impeller arranged in two opposing groups, on several occasions to keep the minimum flow at a desired limit, the discharge temperature rise was permitted up to 50 deg F.

4. Balancing Drum

When a boiler feed pump is equipped with a balancing drum or disk the minimum flow should be maintained to prevent flashing in the balancing chamber beyond the drum (point (4), Fig. 2). Although flashing in the balancing chamber has no bearing on the pump flow or its cavitation characteristics, excessive flashing could possibly produce mechanical vibration.

The minimum flow depends on the available NPSH at the pump suction. For the pump on Fig. 1, NPSH = 72.0 ft. This is equivalent to 28.2 psi at the density prevailing at the suction. Thus the pressure in the balancing chamber is equal to 104.5 + 28.2 = 132.7 psia. Here 104.5 is the saturation pressure at the suction temperature of 331 F. The saturation tempera-
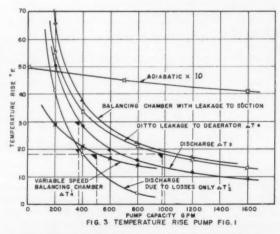
Fig. 3—Temperature rise for the pump of Fig. 1 is to be held within set limits. Curves above help determine allowable minimum flow

ture corresponding to the 132.7 psia is 394 F. This means a temperature rise of 18 deg F is possible in the balancing chamber without flashing. Fig. 3 indicates a minimum flow of 980 gpm for 18 deg F temperature rise. However, this particular pump is equipped with a hydraulic coupling permitting flow control by speed variation. At reduced speeds the power input to water (w) is reduced and thus the temperature rise will be lower than the constant speed curve (ΔT_4) shows. The variable speed temperature curve ($\Delta T_4'$) for constant head is also shown in Fig. 3. This shows a minimum flow of 500 gpm for a temperature rise of 18 deg F.

Actually the minimum flow will be appreciably lower as the pressure in the balance chamber is higher than at the pump suction by the value of the pipe friction loss in the suction pipe from the deaerator to the suction nozzle plus the pipe loss from the deaerator to the balancing chamber.

The minimum flow to suppress the flashing in the balancing chamber can be adjusted to a desired amount by raising the pressure in the chamber by means of an orifice in the leakage pipe to the deaerator.

5. Drum Leakage Piped to Pump Suction

When the drum leakage is piped to the pump suction the inlet water temperature and vapor saturation pressure are increased. The available NPSH value is reduced by the amount of the vapor pressure increase. However, the minimum flow still will be determined by the conditions existing in the balance chamber beyond the drum because this chamber is at the same pressure and NPSH as the pump suction and the temperature there is higher. In the quoted example of pump in Fig. 1 at 200 gpm the temperature in the balance chamber is $T_4 = 386.5$ F (Fig. 2) and with the leakage piped to the suction the temperature of the mixture will be 340 F. Thus temperature of the inlet water is raised by 9 deg F.

With drum leakage piped to the pump suction the temperature rise at the balancing chamber is greater than

with leakage drained to the deaerator because the power input per pound is higher (Fig. 3). In the first case the power input is applied to the $W_d = W_s$ weight of flow ($w = 70$ Btu/hr. at 200 gpm) while in the second case the power input is absorbed by $W_s = W_d + W_L$ weight of flow ($w = 58.5$).

In practice the drum leakage is piped to the deaerating heater. When there is a booster pump ahead of the boiler feed pump the balance chamber is connected with the pump suction to maintain the axial thrust balance. But with such an arrangement there is always ample NPSH provided so that the increase of inlet water temperature does not harm performance.

Conclusions

1. The water temperature rise from suction to discharge in a boiler feed pump is determined by the enthalpy increase, which is equal to the total power input, and not only by power due to hydraulic internal losses.

2. The maximum temperature rise limit based on the internal losses only of 20 deg F, frequently quoted, may correspond to 30 deg or more actual temperature rise based on the enthalpy increase.

3. The temperature rise at the pump discharge has no effect on the suction conditions of the pump as to cavitation, i.e., NPSH available and required are independent of discharge temperature.

4. The minimum flow for pumps provided with thrust balancing means is determined by the temperature rise in the balancing chamber irrespective of whether the drum leakage is piped to the pump suction or deaerator. However, the temperature rise curve for the case of leakage piped to the pump suction is slightly higher than in the case of leakage piped to the deaerator.

5. The minimum flow for pumps without thrust balancing devices is set by the assumed temperature rise at pump discharge. A temperature rise of 30 deg F based on the enthalpy rise should be considered as conservative There are installations with 50 deg F or more temperature rise in successful operation.

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8. Sack "*Uber die thermodynamische Methode zur Bestimmung der Wellen leistung und des Wirkangsgrades von Hoehdruck Speisepumpen*" Brennstoff-Wärme-Kraft (BWK), 1959, No. 11, pp. 511–517.

Construction Begun on Sea Water Conversion Plant

The Westinghouse Electric Corp. has begun construction of a one-million-gallon per day plant that will convert sea water to fresh water at Point Loma near San Diego, Calif. Once in operation, it will be the largest multistage flash-evaporator plant in the United States.

The flash-evaporation process used in the Point Loma plant will consist essentially of spraying heated sea

water under pressure into a chamber that is at a lower pressure and temperature. A portion of the water "flashes" into vapor and is then condensed, providing water that is nearly free of impurities. The remaining salty water passes through a series of additional chambers where the flashing process is repeated. At each additional stage the condensed, salt-free water is piped off.

(Drawn from the Monthly Technical Bulletin, International Combustion, Ltd., London, W. C. 1)

Fuels: Sources, Properties and Preparation

Getting the Best out of Coal. Anon. *Steam Engr.* 1960, **30** (Dec.), 75–80.

The papers presented at the Conference of the Combustion Engineering Conference at Harrogate, November 8 and 9, 1960 and the discussions are reviewed. These included: (1) Choosing the right coal by A. H. Slade; (2) Sampling of coal by R. C. Tomlinson; (3) Operating and Maintaining the Plant by F. L. Griffiths; (4) Choosing Plant, one by J. E. H. Moore and one by R. A. Kirby; (5) New Developments by G. G. Thurlow.

Coal to Make 750,000 kw. D. H. Kregg. *Pwr. Engng.* 1960, **64** (Nov.) 82–3.

The coal handling plant at the Thos. H. Allen power station on the bank of the Mississippi consists of twin unloading towers with a capacity of 1500 t/h each, a hopper of 900 t capacity, dual belts to the transfer tower, a traveling conveyor for stacking out, two reclaiming hoppers and conveyors to the crusher house equipped with two crushers of 1200 t/h total capacity and two conveyors for filling the bunkers on opposite sides of the boilers (opposed cyclone fired). The control room is installed in the transfer tower containing a mimic bus and lights and bunker level indicators.

Coal Sampler Gets to Bottom of Heap. Anon. *Elect. Wrld.* 1960, **154** (Nov. 28), 57.

The sampler for taking representative samples from a loaded wagon consists of a hollow tube pneumatically driven to a predetermined depth when two hinged leaves close. After withdrawal of the tube the hinged leaves open, the sample is inspected visually and then sent to the laboratory.

Steam Generation and Power Production

Hydraulic Instability in a Natural Circulation Loop with Net Steam Generation at 1000 Psia. S. Levy and E. S. Beckjord. *ASME Preprint* No. 60-HT-27 1960 (Aug.), 20 pp.

The causes of instability and the freqency of the oscillations were studied. It is not yet possible to obtain a complete understanding of the phenomena from the test data.

Natural-Circulation Tests with Water at 800 to 2000 psia under Nonboiling, Local Boiling and Bulk Boiling Conditions. O. J. Mendler, A. S. Rathbun, N. E. Van Huff and A. Weiss. *ASME Preprint* No. 60-HT-36 1960 (Aug.), 17 pp.

Natural and forced circulation test data for a closed loop system are presented. No differences in results were found for forced and natural circulation. Flow fluctuations during natural circulation were observed before burnout heat flux was reached causing premature burnout in some cases.

A Variable Density Single-Fluid Model for Two-Phase Flow with Particular Reference to Steam-Water Flow S. G. Bankoff. *J. Heat Transf.* 1960, **82** (Nov.), 265–72.

Two equations have been derived making possible the prediction of the mean void fraction and frictional pressure drop for the flow of a steam-water mixture of known quality at a given pressure.

Influence of Different Fuels on Boiler Design. O. Engler. *Energie* 1960, **12** (Nov.), 482–6 (in German).

The following are illustrated and briefly described: (1) Benson boiler rated at 1100 klb/h at 2418 psi and 985/985 F, for brown coal with 15 per cent sand, roof fired, with horizontal low-level convection pass containing pendant superheater, reheater and economiser, followed by horizontal air preheater and precipitator; (2) New mixing burner with tangential coal and air inlets resulting in complete combustion with only traces of CO in the furnace; (3) Boiler with slag-tap furnace and opposed burners, tangent tubes with attached insulation and boiler casing; (4) Russian boiler insulation and casing attached to tube panels before erection; (5) Boilers with circular grate or inclined (Möring) stoker for coke firing; (6) Boiler for gas and oil firing with spring-steel band air preheater behind Ljungstrom preheater permitting cooling of flue gas below few point without corrosion; (7) Boiler for oil and sulphite lye; (8) Boiler with Martin stoker for refuse burning and slag-tap furnace for coal; (9) Boilers for blast furnace gas and oil or coal.

Oil Fired Corner-Tube Boilers. H. Oestreich. *Energie* 1960, **12** (Nov.), 494–8 (in German).

Examples of oil fired corner-tube boilers with and without superheater and of different size are illustrated. Burners may be installed in the roof, front wall or bottom of the furnace. The difficulties of designing boilers for alternative firing of coal and oil are discussed.

Superheater Supports Made of 40-60 Nickel Chromium Alloy Withstood Corrosion from Marine Fuel Oil Ash for Over 7000 h. Anon. *Chem. Proc.* 1960, **6** (Dec.), 6–7.

Superheater supports of 25 per cent Cr-12 per cent Ni iron base alloy had an effective life of 3000–4000 h while those made of a 60 per cent Cr-40 per cent Ni alloy showed little wastage after 7000 h and have a service life expectancy of 17,000 h.

New Chemical Cleaning Concepts and Possible Applications for the Future. C. M. Louks. *ASME Preprint* No. 60-WA-259 1960 (Dec.), 4 pp.

Chemical cleaning methods of boilers at present in use are discussed and possible new methods eliminating time-consuming processes and permitting cleaning of superheaters, reheaters and steam lines by non-condensed or vapor phases are indicated.

Chemical Cleaning with Citric Acid Solutions. W. E. Bell. *ASME Preprint* No. 60-WA-257 1960 (Dec.), 3 pp.

The removal of deposits such as scales and oxides by ammoniated 3 per cent citric acid solutions (pH 3–3.5) instead of citric acid is discussed; by using the ammoniated solution no ferrous acid citrate precipitation occurs as it does with citric acid.

Solid Fuel Firing

Progress in the Design of Mechanical Stokers. J. P .Lauder. *Engng. Boil. Ho. Rev.* 1960, **75** (Dec.), 399–404.

Recent improvements on mechanical stokers are described. In double-flue boilers with low-ram stoker a supplementary gear box has been added to one of the feeders which in conjunction with variable throw crank magnifies or reduces the movement of the other, or two motors are used to drive the feed mechanism and grate bars independently; grate bars are now made of chromium steel to B.S.S. 1648–1950 Grade "C"; a new side sealing plate prevents accumulation of clinker; a new ram with increased travel improves feeding of wet and small coal from the vertical-wall hopper; a newly designed screw-feeder for moving coal from the ground to bunker or hopper is described. For chain-grate stokers two methods of ensuring high-pressure combustion air being admitted at the front and low-pressure air at the rear by the "dual-zone" or "contraflow" stoker are outlined. A new control is applied to underfeed stokers to prevent smoke emission on starting and stopping; the air fan continues running for a while after the coal supply has been stopped and is started before feeding in green coal.

Liquid and Gaseous Fuel Firing

Influence of Combustion Conditions on the Dew Point of Flue Gases of Oil Fired Boilers. K. Cleve. *Energie* 1960, **12** (Nov.), 486–90 (in German).

Based on laboratory and field tests on boilers of different size and type of burners the relationships between dew point and excess air, load, primary air admission and uniform flow of oil to the burners are discussed. It is concluded that the minimum flue gas temperature is 250 F without additives but should preferably be 280–300 F. Minimum excess air and maximum CO_2 content of about 15 per cent should be aimed at.

New Insight into the Combustion of Sulphur Containing Fuel Oil. W. Hoffmann. *Energie* 1960, **12** (Nov.), 491–4 (in German).

Recent investigations by the author and others lead to these conclusions: (1) The SO_3 content of the flue gas does not depend on the S content of the oil, it depends on combustion conditions and all factors influencing combustion; coarse atomization with good mixing of air produces less SO_3 than fine atomization with bad mixing; (2) The term sulfuric acid dew point should be replaced by beginning sulfuric acid deposition; (3) Corrosiveness depends on the amount of sulfuric acid formed during combustion; no relationship exists between sulfuric acid content of the flue gas and S content of the oil; (4) Chimney emission of smut and smoke is a problem of burner and furnace design.

Furnaces and Combustion

The Influence of Volatile Matter on the Combustion of Pulverized Coal. J. F. Mullen and G. Gould. *ASME Preprint* No. 60-FU-3 1960 (Oct.), 4 pp.

The evolution, composition and ignition limits of volatile matter in coals are considered in general and their effect on the combustion of pulverized coals of different V.M. content in particular. The appearance of fish tails, blow back, pulsating burners and high carbon losses is discussed as a function of the release and ignition of volatile matter and combustion conditions.

The Cyclone Furnace. A. G. Roberts. *B.C.U.R.A. Monthly Bull.* 1960, **24** (Apr./May), Pt. II, 205–54.

This review deals with: (1) Main types of cyclone furnaces used for firing boilers; (2) Size of cyclone furnaces and cyclone-fired boilers; (3) Commercial developments; (4) Coal preparation and feeding; (5) Removal and disposal of slag; dust emission; (6) Suitability of coals; (7) Use of fuels other than coal; (8) Power requirements for cyclone-fired boilers; (9) Maintenance and availability; (10) Safety; (11) Combustion control; (12) Corrosion, erosion and deposition; (13) Cyclone furnaces for gas turbine cycles; (14) Other designs and applications of the cyclone furnace; (15) Combustion, heat transfer and pressure drop in the cyclone furnace; (16) The present position of cyclone firing.

Ten Years of Operating Experience of Opposed Burner Slag-tap Furnaces. K. Nuber. *Energie* 1960, **12** (Nov.), 479–82 (in German).

Experience has shown that in boilers with this type of slagging furnace corrosion, erosion or deposit formation has rarely occurred, especially since the introduction of a mixing

burner which has proved equally suitable for low and high volatile (5-50 per cent) coals. Soot blowers and steel shot cleaners were not required and have often been removed. Partial load down to 30 per cent can be carried with liquid slag removal. Ash retention is deliberately limited to 50–65 per cent, the remaining fly ash being used to keep the boiler clean.

Water-Side Corrosion and Water Treatment

Internal Corrosion of Power Station Boilers. E. C. Potter. *Br. Pwr. Engng.* 1960, **1** (Dec.), 31–3.

A general survey of causes of corrosion such as caustic cracking and oxygen pitting, failure of protective film and contributory factors, of methods of detection of corrosion and means to ensure freedom from corrosion.

Twenty-First Annual Water Conference. Anon. *Combustion* 1960, **32** (Nov.), 46–50.

Brief abstracts of papers and discussions on different aspects of feed water and condensate treatment and chemical cleaning of boilers and nuclear plants are presented.

The Conditioning and Control of Feed Water in the High Pressure Power Stations of Électricité de France. R. Rath. *Chal. et Ind.* 1960, **41** (Nov.), 309–17 (in French).

The amounts of trisodium phosphate and morpholine added to the boiler feed water for boilers operating at different pressures are tabulated and compared with American and British practice. Automatic control of the feed water is supplemented by regular manual control.

The Control and Measurement of Hydrazine in Thermal Power Stations. R. Massart and L. Missa. *Centre Belge d'Etude et de Doc. des Eaux Bull.* Trim. 1960, **III** 162–70 (in French).

The paper deals with the deterioration of hydrazine, effect of the nature of the salts in the water, the part played by oxygen in air, prevention of deterioration and dissociation of hydrazine in a boiler, with reference to the results, given in the form of graphs, of observations on these matters carried out at Awirs power station, Belgium. *C.E.G.B. Digest* 1961, **13** (Jan. 21), 189.

Gas-Side Corrosion and Deposits

Effect of Ammonia Injection on Corrosion in Air Preheaters. R. Kato and B. E. Paris. *ASME Preprint* No. 60-WA-255 1960, (Dec.), 7 pp.

The first trials with ammonia injection into the flue gas of a pulverized coal fired boiler (320 klb/h, 840 psi, 850 F) at B.E. Morrow station are reported. The gas inlet and exit temperatures are 420 and 310 F, those of the air 105 and 280 F. The coal has an S content of 3.9–4.2 per cent. A constant temperature corrosion probe was used. By the use of ammonia at the rate of 0.06 per cent by weight of fuel corrosion at metal temperatures between 150 and 420 F was eliminated, but not below 150 F.

Flue Gas, Ash and Dust

Pulverized-Fuel Ash as a Structural Material. A. E. Williams. *Engng. Boil. Ho. Rev.* 1960, **75** (Dec.), 407–9.

Some examples of the use of fly ash replacing sand in concrete structures, in road construction, dams and building foundations are described.

Measures for the Reduction of Dust Emission in an Older Industrial Power Station. F. Bellendorf. *Energie* 1960, **12** (Nov.), 499–502 (in German).

In a power station built in 1938/41 containing 10 boilers generating a total of 1800 klb/h at 1850 psi and 932 F originally equipped with dry bottom but later converted to slag-tap furnaces the chimney (150 ft height) emitted excessive amounts of dust. Experiments were made with the installation of multi-cyclone separators before and behind the existing electrostatic precipitator. Although the tests showed that in this particular case the installation of the separator behind the precipitator gave better results and was easier to adopt because of the general layout of the plant it is concluded that this need not always be the case; only if the dust is very fine does this arrangement appear to be more successful.

Heat Recovery Plant

Thin Tube Plates for H.P. Heat Exchangers. Pt.I and II. D. E. Hartley. *Nucl. Engng.* 1960, **5** (Dec.), 557–60, 1961, **6** (Jan.), 8–11.

Although theoretical predictions would allow thin tube plates to be used it has been found in experimental investigations that it is in fact possible to do so without exceeding stress limits. Experimental evidence is presented which shows that thin tube plates are in fact stronger than predicted by theory.

Shell-and-Tube Heat Exchangers: Effect of By-Pass and Clearance Streams on the Main Stream Temperature. B. E. Short. *ASME Preprint* No. 60-HT-16 1960 (Aug.), 8 pp.

By-pass is the space between the tubes and the shell and clearance the space between the baffles and the shell. The mixing of the streams flowing through these spaces with the main stream and the resultant temperatures of the mixed and the main stream are considered in their effect on heat transfer coefficient and pressure drop.

Performance of Stainless-Steel Condenser Tubes. R. H. Pell. *ASME Preprint* No. 60-WA-217 1960 (Dec.), 9 pp.

The excellent operating experience is described. Advantages are better heat transfer than expected, high corrosion and fouling resistance, tendency to self-cleaning and scouring, reduced erosion and wear.

Power Generation and Power Plant

Critical and Supercritical Steam Pressures. P. B. Silk. *Pwr. and Wrks. Engng.* 1960, **55** (Dec.), 939–42.

Subcritical, critical and supercritical steam cycles with and without reheat are compared and the gain in thermal efficiency, especially in large units, shown.

More Economical New Power Stations. K. Schröder. *Elektwirt* 1960, **59** (Nov. 20), 786–91 (in German).

Present preferred unit size is 100–250 mw with steam at 2400–2700 psi and 986–1050 F. Possibilities of reducing overall generating costs are discussed under: (1) Reduced specific heat consumption; (2) Decrease of specific construction costs; (3) Savings in operational costs; (4) Favorable purchases of fuel. It is suggested that installation of supercritical pressure units would materially improve specific heat consumption and allow larger units to be constructed. Reduced capital costs would result from outdoor installation. It is not believed that increased automatic control would lead to a reduction of personnel.

A Decade of Electric Utility Fuel Experience. M. E. Robinson and W. L. Kurtz. *ASME Preprint No.* 60-FU-1 1960 (Oct.), 13 pp.

Statistical data are provided on the consumption and efficiency of consumption of coal, oil and gas in the various regions of the U.S.A. between 1949 and 1958. It is shown that the efficiency of burning coal expressed by Btu/kwh has increased more rapidly than that of oil and gas.

6th Steam Station Design Survey L. A. Allen. *Elect. Wrld.* 1960, **154** (Nov. 21), 73–92.

Details of 72 new plants are tabulated. The trend is towards increase in size, temperature and pressure.

Increased automatic control by computer is noted. As regards boilers divided furnaces are increasing, twin furnaces decreasing, pressurized firing of coal fired plants is increasing and also gas recirculation for superheat control. No supercritical pressure unit is planned.

System Expansion Fuel Costs Versus Capital. W. J. Fahrner and C. M. Heidel. *ASME Preprint* No. 60-WA-156 1960 (Dec.), 5 pp.

The procedure instituted by Detroit Edison Co. for making decisions on future plant installation is based on a computer-type programme providing short-range and long-range system plans.

Thermal Stress Protection in Starting and Loading Boiler-Turbine-Generator Combinations. R. H. Reisinger and C. B. Scharp. *ASME Preprint* No. 60-PWR-3 1960 (Sept.), 8 pp.

Procedures developed for starting boiler-turbine units from cold and hot have been worked out to avoid thermal shock cracking.

Heat and Power Economy in Denmark. H. Reimer. *Energie* 1960, **12** (Nov.), 459–72 (in German).
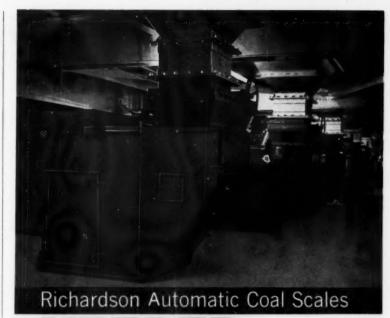
Denmark has 6 heat-power stations supplying electric energy to the towns and surrounding districts and heat (hot-water) to consumers in the towns; 60 stations supply heat only to their neighborhoods. Condensing power stations work in close collaboration with the 6 heat-power stations. These 6 stations, pipe lines, cost division between heat and power, house connections and costing methods for heat are described.

Energy Economy in Iron and Steel Works. R. Simon. *Energie* 1960, **12** (Nov.), 472–8 (in German).

The development of energy-consumption in different types of iron and steel works is traced, supply of energy to and from the works and losses, supply of blast furnace and coke-oven gas to various consumers is discussed and generation and consumption of electric power and preheated air described. Waste heat utilization, especially generation of steam has increased considerably in recent years.

Ventilation of Eddystone Station—An Approach to Ventilation of Modern Steam-Electric Generating Stations. S. J. Kowalski. *ASME Preprint* No. 60-WA-44 1960 (Dec.), 8 pp.

It is suggested that ventilation of a boiler room should result in a heat loss of 1 per cent of coal burned with a temperature rise of 20–25 F and a turbine room 0.9 per cent of generator name plate rating with a rise of 15–20 F.

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